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PATENT & TRADEMARK OFFICE

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Patent and Trademark Office, U.S. DEPARTMENT OF COMMERCE

REQUEST FOR REEXAMINATION TRANSMITTAL FORM

TO:
Mail Stop *Ex Parte Reexam*
Commissioner for Patents
P.O. Box 1450
Aexandria, VA 22313-1450

Attorney Docket No. 8351.0294
Attorney Customer Number: 22,852
Date: January 20, 2006

1. ☒ This is a request for *ex parte* reexamination pursuant to 37 CFR 1.510 of patent number 6,688,280 issued to Weber et al.
   The request is made by: ☒ patent owner. ☐ third party requester.
2. ☒ The name and address of the person requesting reexamination is:
   Caterpillar Inc.
   100 N.E. Adams Street
   Peoria, IL
3. ☒ a. A check in the amount of $2,520.00 is enclosed to cover the reexamination fee, 37 CFR 1.20(c)(i);

   ☒ b. The Commissioner is hereby authorized to charge any missing or additional fees as set forth in 37 CFR 1.20(c) to Deposit Account No. 06-0916.
4. ☒ Any refund should be made by ☐ check or by ☒ credit to Deposit Account No. 06-0916. 37 CFR 1.26(c)
5. ☒ A copy of the patent to be reexamined having a double column format on one side of a separate paper is enclosed. 37 CFR 1.510(b)(4)
6. ☐ CD-ROM or CD-R in duplicate, Computer Program (Appendix) or large table
7. ☐ Nucleotide and/or Amino Acid Sequence Submission: *If applicable, all of the following are necessary*
   a. ☐ Computer Readable Form (CRF):
   b. Specification Sequence Listing on:
      i ☐ CD-ROM (2 copies) or CD-R (w copies); **or**
      ii ☐ paper
   c. ☐ Statements verifying identity of above copies
8. ☐ A copy of any disclaimer, certificate of correction or reexamination certificate issued in the patent is included.
9. ☒ Reexamination of claim(s) 1-16 and 21-33 is requested.
10. ☒ A copy of every patent or printed publication relied upon is submitted herewith including a listing thereof on Form PTO SB/08. Also enclosed is an Information Disclosure Statement, along with an additional Form PTO SB/08, copies of each of the listed foreign patent and non-patent literature documents, and each of the English language abstracts and translations identified on the additional form.

11. ☒ An English language translation or abstract for non-English language patents or printed publications is included.

12. ☒ The attached detailed request includes at least the following items:
   a. A statement identifying each substantial new question of patentability based on prior patents and printed publication. 37 CFR 1.510(b)(1)
   b. An identification of every claim for which reexamination is requested, and a detailed explanation of the pertinency and manner of applying the cited prior art to every claim for which reexamination is requested. 37 CFR 1.510(b)(2)

13. ☐ A proposed amendment is included (only where the patent owner is the applicant). 37 CFR 1.510(e)

14. ☐ a. It is certified that a copy of this request (if filed by other than the patent owner) has been served in its entirety on the patent owner as provided in 37 CFR 1.33(c). The name and address of the party served and the date of service are:
[Name]
[Address]
[Address]
Date of Service [Date]; **or**

☐ b. A duplicate copy is enclosed since service was not possible.

15. ☒ Correspondence Address: Direct all communication about the reexamination to:
Finnegan, Henderson, Farabow, Garrett & Dunner, L.L.P.
Customer Number: 22,852

16. The patent is currently the subject of the following concurrent proceeding(s):
   a. Copending reissue application Serial No. [Text].
   b. Copending inter partes reexamination Control No. 95/000,050.
   c. Copending Interference No. [Text]
   d. Copending litigation styled: [Text]

Anthony M. Gutowski
Reg. No. 38,742
Finnegan, Henderson, Farabow,
Garrett & Dunner, LLP
Dated: January 20, 2006

☒ For Patent Owner Requester

☐ For Third Party Requester

PATENT
Customer No. 22,852
Attorney Docket No. 08351.0294-00000

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

In re Reexamination of:	)	
U.S. Patent No. 6,688,280 to Weber et al.	)	Group Art Unit: To Be Assigned
Issue Date: February 10, 2004	)	Examiner: To Be Assigned
Reexam Control No.: To Be Assigned	)	
Filed: January 20, 2006	)	
For: AIR AND FUEL SUPPLY SYSTEM FOR COMBUSTION ENGINE	)	

REQUEST FOR *EX PARTE* REEXAMINATION

Mail Stop *Ex Parte* Reexam
Hon. Commissioner for Patents
P.O. Box 1450
Alexandria, VA 22313-1450

Sir:

The following is a Table of Contents for this Request for *Ex Parte* Reexamination:

I. INTRODUCTION

Caterpillar Inc. (hereinafter the "Requester") requests reexamination under 35 U.S.C. § 302 and 37 C.F.R. § 1.510 of U.S. Patent No. 6,688,280 ('280 patent), issued February 10, 2004, to inventors James R. Weber and Scott A. Leman. The '280 patent is assigned in its entirety to the Requester.

A. NOTICE OF CONCURRENT *INTER PARTES* REEXAMINATION CONTROL NO. 95/000,050

Pursuant to 37 C.F.R. § 1.565, the Requester hereby provides notice of a concurrent *inter partes* reexamination proceeding, Reexamination Control No. 95/000,050, involving U.S. Patent No. 6,688,280, the same patent at issue in this *ex parte* reexamination request. The concurrent *inter partes* reexamination was filed by a third party requester on September 17, 2004, based on a primary reference to Clyde C. Bryant, International Application No. WO 98/02653 ("Bryant").

The primary reference of this *ex parte* Reexamination Request is a prior art publication to G. Zappa et al. ("Zappa") entitled "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio." Zappa was not cited during the original prosecution of the '280 patent. Further, Zappa has not been considered in the pending *inter partes* reexamination proceeding for the '280 patent. More importantly, as discussed in the attached Declaration of Dr. Joel Hiltner at paragraphs 6-7, Zappa is much more relevant to the '280 patent than Bryant, the primary reference applied in the pending *inter partes* reexam. For example, Bryant provides only a scattered discussion setting forth numerous listings of alternative features without providing any meaningful disclosure of how to combine particular

engine features, such as any particular intake valve timing, fuel supply, air compression/non-compression, etc. Additionally, Bryant lacks a disclosure of turbochargers and variable compression ratios.

In contrast to Bryant's catalog of unconnected features, Zappa discloses several engine aspects that are directly linked to one another. In particular, Zappa provides specific disclosure of engines and methods of engine operation that include both an intake valve closing extremely late in the compression stroke (e.g., after a majority portion of the compression stroke) and fuel being injected directly into a combustion chamber during both the compression and expansion strokes, after the intake valve closes. No such disclosure of late intake valve closing in combination with fuel injection timing is contained in Bryant. Also, in contrast to the absence of disclosure of turbochargers and variable compression ratio in Bryant, Zappa discloses both series turbochargers (e.g., at page D19-3, first - third paragraphs, and page D19-13, Fig. 1) and variable compression ratio (e.g., at page D19-2, lines 22-25, referring to "main features" including "variable compression ratio," and page D19-3, fifth - eighth paragraphs describing "Variable Compression Ratio").

B. REQUIREMENTS OF REEXAMINATION REQUEST

This Request is accompanied by the following:

(1) The appropriate fee payment of $2,520 under 37 C.F.R. § 1.20(c);

(2) A statement pointing out each substantial new question of patentability based on the newly cited prior art;

(3) An identification of every claim for which reexamination is requested, and a detailed explanation of the pertinency and manner of applying the cited prior art to every claim for which reexamination is requested;

(4) A copy of each of the newly cited prior art references discussed in this Request and a listing thereof on a Form PTO SB/08, along with an Information Disclosure Statement and a second SB/08 form listing additional references for consideration by the Examiner; and

(5) A copy of the entire patent including the front face, drawings, and specification/claims, in double column format.

Because the Requester is the owner of the entire interest in the patent for which reexamination is requested, no service on the patent owner is necessary.

II. CLAIMS FOR WHICH REEXAMINATION IS REQUESTED

Reexamination is requested for claims 1-16 and 21-33 of the '280 patent. Of these, claims 1, 7, 13, 21, and 29 are in independent form.

III. APPLICATION OF THE CITED PRIOR ART TO THE CLAIMS

- Requester seeks reexamination of claims 1-12, 21, 22, and 25-30 of the '280 patent in view of Zappa et al., A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio; CIMAC, 1979, alone.
- Requester seeks reexamination of claims 13-16 of the '280 patent in view of Zappa alone or in combination with U.S. Patent No. 5,445,128 to Letang et al. ("Letang").
- Requester seeks reexamination of claims 23, 24, and 31-33 of the '280 patent in view of Zappa alone or in combination with U.S. Patent No. 4,836,161 to Abthoff et al. ("Abthoff").

The relevance of Zappa to the claims of the '280 patent is discussed in the attached claim chart entitled "Comparison of '280 Patent Claims to Zappa et al." and in

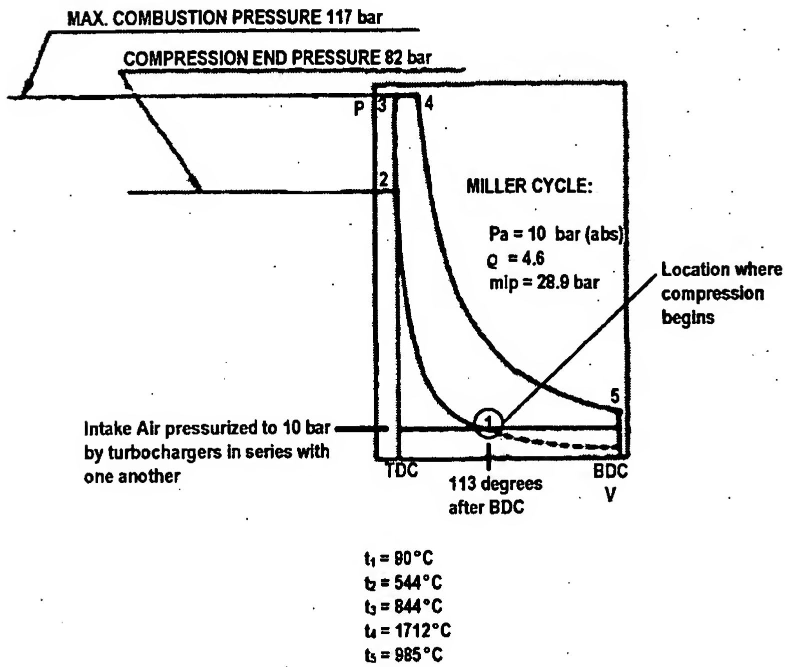
the attached Declaration of Dr. Joel Hiltner. In addition, the following also provides a discussion of the relevance of Zappa, Letang, and Abthoff to the claims.

A. Zappa

Zappa discloses a four-stroke, diesel, internal combustion engine having a two-stage supercharging system involving a pair of turbochargers in series with one another, as well as a first intercooler positioned between the pair of turbochargers ("low pressure cooler") and a second intercooler located downstream of the turbocharger pair ("high pressure cooler"). (See page D19-3 of Zappa.) Fig. 6 of Zappa disclose an intake valve being closed very late in the compression stroke at 113 degrees after bottom dead center of the compression stroke, and that figure also discloses closing an air intake valve very early in the intake stroke at 113 degrees before bottom dead center of the intake stroke. See, Hiltner Declaration at paragraphs 11-16. In Zappa, Figs. 6 and 8 disclose an air intake valve being selectively operated to allow pressurized air to flow between a combustion chamber and an intake manifold during at least a portion of an intake stroke and through a majority of a compression stroke. See, Hiltner Declaration at paragraphs 11-21.

In particular, as shown below in an annotated version of the far right diagram included in the diagrams of Fig. 6 of Zappa, the location 1 of the line 1-2-3-4-5 shows the beginning of in-cylinder pressure increase caused by upward piston movement, and location 2 is the end of the compression stroke (TDC of the compression stroke).

Annotated Fig. 6 from Zappa et al.

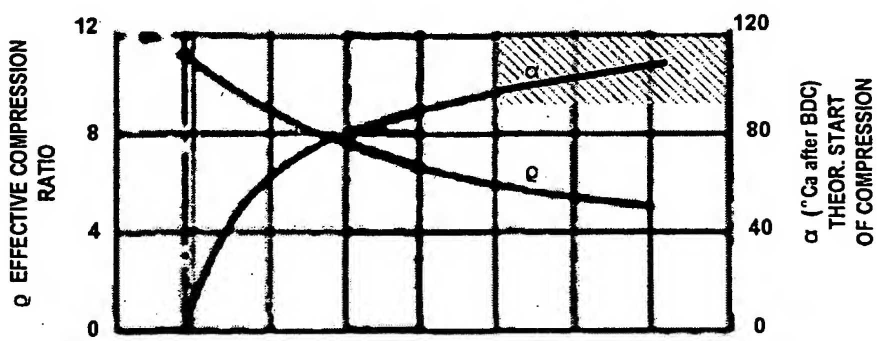


The distance along the X-axis from the location 1 to the piston TDC volume is shorter than the distance along the X-axis to the piston BDC volume. Such a relative spacing of location 1 discloses an intake valve being held open through more than 50% of a compression stroke (i.e., through a majority portion of the compression stroke) and it also discloses an alternative arrangement involving an intake valve being held open for only less than 50% of an intake stroke (i.e., through a minority of the intake stroke). See, Hiltner Declaration at paragraphs 13-14. Stated another way, the position of location 1 discloses an intake valve being closed more than 90 crank angle degrees after bottom dead center of the compression stroke and less than 90 crank angle degrees after top dead center of the intake stroke. Indeed, according to Dr. Hiltner, the far right diagram of Fig. 6 shows both an intake valve closing very late at 113 crank

angle degrees after bottom dead center of the compression stroke, and an intake valve closing very early at 113 crank angle degrees before bottom dead center of the intake stroke. See, Hiltner Declaration at paragraph 16.

Fig. 8 of Zappa and a separate discussion of "very delayed closing" at page D19-7 of Zappa are consistent with this disclosure of Fig. 6. For example, the following figure is an annotated version of Fig. 8 of Zappa, including a curve α representing the theoretical start of engine cylinder compression in terms of crank angle degrees after bottom dead center of the compression stroke.

Annotated Fig. 8 from Zappa et al.



As reflected by the portion of curve α passing into the cross-hatched portion of annotated Fig. 8, curve α shows cylinder compression starting more than 90 degrees after bottom dead center of the compression stroke. Therefore, Fig. 8 of Zappa discloses an intake valve being closed more than 90 degrees after bottom dead center of the compression stroke and also discloses an intake valve being closed less than 90 degrees after top dead center of the intake stroke. See, Hiltner Declaration at paragraph 20.

In addition, the far right chart of Fig. 6 of Zappa also discloses fuel being injected directly into a combustion chamber after an intake valve closes and during at least a portion of the compression and expansion strokes of the piston. In particular, line segment 2-3 shown above represents an increase in cylinder pressure without any substantial change in cylinder volume above a piston, and location 3 is at a pressure that Zappa refers to as the maximum "combustion pressure." In light of the fact that Zappa refers to direct injection of diesel fuel (for example, at page D19-10 in the last paragraph), one of ordinary skill in the art would understand that the increase in pressure illustrated by line segment 2-3 is caused by fuel being directly injected into the cylinder during the compression stroke after the intake valve closes at location 1. See, Hiltner Declaration at paragraph 23.

Line segment 3-4 of the far right diagram of Fig. 3 represents a relatively constant "combustion pressure" in the cylinder while the cylinder volume above the piston increases due to the piston moving from top dead center in the expansion stroke. The far right diagram of Fig. 6 also shows an increase in temperature from 844 degrees C to 1712 degrees C during the constant combustion pressure of line segment 3-4. One of ordinary skill in the art would understand that line segment 3-4 and the temperature increase illustrate a constant pressure phase of combustion in the cylinder caused by fuel being injected into the combustion chamber after top dead center during the expansion stroke. See, Hiltner Declaration at paragraph 24.

The far right chart of Fig. 6 of Zappa shows conventional diesel combustion involving both a constant volume phase (illustrated by line segment 2-3) and a constant pressure phase (illustrated by line segment 3-4), which require direct fuel injection at the

end portion of the compression stroke and at the beginning of the expansion stroke. As discussed in the Hiltner Declaration at paragraph 25, Obert, International Combustion Engines, *Analysis and Practice*, Second ed., 1950, at pgs. 142-144 and 153-154 (Exhibit E of the Hiltner Declaration) confirms that the far right diagram of Fig. 6 of Zappa discloses fuel being injected at the end portion of the compression stroke and at the beginning of the expansion stroke.

In view of the above, Zappa discloses a 4-stroke internal combustion engine including the following:

- a pair of turbochargers in series;
- a first intake intercooler between the turbochargers;
- a second intake intercooler downstream of the turbochargers;
- an inlet valve control device varying the closing timing of the intake valve(s);
- closing the intake valve(s) after a majority portion of the compression stroke; and
- injecting fuel into the combustion chamber after closing of the intake valve(s) and during the compression and expansion strokes.

Requester refers to the attached claim chart and Declaration of Dr. Joel Hiltner to provide a further explanation of why Zappa is directly relevant to the subject matter of claims 1-16 and 21-33 of the '280 patent. In view of the disclosure of Zappa, Requester submits that Zappa is directly relevant to claims 1-16 and 21-33 of the '280 patent.

B. Letang

Letang discloses an internal combustion engine having electronic controls for various engine functions, and states that it is desirable to use a comprehensive controller integrating the various engine functions. The use of a comprehensive controller would, for example, reduce part proliferation and cost. (See col. 3, lines 15-24 of Letang.) In view of the disclosure of Letang, Requester submits that Letang may be relevant to one or more of claims 13-16 of the '280 patent that recite, *inter alia*, a controller configured to selectively operate an air intake valve, wherein the controller is configured to inject fuel.

C. Abthoff

Abthoff relates to a direct fuel injection method for a diesel engine in which a preset quantity of fuel is introduced into a combustion space via a fuel injection nozzle in a preinjection and in a main injection which is separate from the preinjection. The beginning of the preinjection occurs within the range of 10° to 16° or 20° to 30° crank angle before top dead center, depending on speed. The beginning of the main injection occurs within the range of 2° after top dead center to 15° before top dead center. (See Fig. 1 of Abthoff.) In view of the disclosure of Abthoff, Requester submits that Abthoff may be relevant to one or more of claims 23, 24, and 31-33 of the '280 patent.

IV. SUBSTANTIAL NEW QUESTION OF PATENTABILITY

Thus, the Requester believes that Zappa, Letang, and Abthoff raise substantial new questions of patentability as to claims 1-16 and 21-33 of the '280 patent.

V. MERGER WITH *INTER PARTES* REEXAM CONTROL NO. 95/000,050

The MPEP recommends merger of a concurrent *ex parte* reexamination with a pending *inter partes* reexamination:

> If a second request for reexamination is filed where a certificate will issue for a first reexamination later than three months from the filing of the second request, the proceedings normally will be merged once reexamination has been ordered in both proceedings.
> (MPEP 2686.01)

Based on the current stage of the *inter partes* reexamination proceeding, the Requester's counsel understands that a certificate should not be issued in the proceeding within the next three months. The Requester respectfully submits that, pursuant to 37 C.F.R. § 1.989 and MPEP 2686.01, this *ex parte* reexamination request should be merged with the pending *inter partes* reexamination immediately upon grant of this *ex parte* request. All of the claims under consideration are identical in both reexamination proceedings. *See* MPEP 2686.01, III.

VI. CONCLUSION

In view of the foregoing, the Requester respectfully requests that the Request for Reexamination be granted.

Respectfully submitted,

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Dated: January 20, 2006

By: ______________________
Anthony M. Gutowski
Reg. No. 38,742

Attorney For Patent Owner/Requester
CATERPILLAR INC.

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	**Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979**
1. A method of operating an internal combustion engine	Zappa *et al.* discusses a "4-Stroke High Speed Diesel Engine," which is an internal combustion engine. *See* page D19-1, title. Page D19-2 provides a general description of the internal combustion engine, and pages D19-13 and 14 provide schematics and a photo of the internal combustion engine.
including at least one cylinder and a piston slidable in the cylinder, the method comprising:	The "geometrical characteristics" described on page D19-2 include a "cylinder" with specified dimensions, and "pistons" are included as "engine components" in the last paragraph on that page. Figs. 15 and 16 on pages D19-20 and 21 illustrate the pistons and cylinders.
supplying pressurized air from an intake manifold to an air intake port of a combustion chamber in the cylinder;	The engine described in Zappa *et al.* includes a "Two-stage Supercharging System" to pressurize air. *See* page D19-3. Fig. 1 on page D19-13 shows an intake manifold after the "two-stage supercharging system," and page D19-6 mentions a "pressure drop between air and gas manifold pressure." The "combustion chamber" is described on page D19-4, ¶1, page D19-5, ¶3, and D19-8, ¶4. The "intake valves" described on pages D19-3, ¶3, D19-6, ¶1, D19-7, penultimate paragraph, and illustrated in Figs. 15 and 16 on pages D19-20 and 21 are valves for the air intake ports into the "combustion chambers" of the cylinders. Figs. 6 and 8 on pages D19-16 show the "supercharging pressure" ("P_a") of the pressurized air supplied to the intake port.
selectively operating an air intake valve to open the air intake port to allow pressurized air to flow between the combustion chamber and the intake manifold	The "intake valves" described at the top of page D19-6 and illustrated in Figs. 15 and 16 on pages D19-20 and 21 are valves for the air intake ports into the "combustion chamber" of the cylinders. Control of the "intake valves" as described on pages D19-3, ¶5, D19-7, penultimate paragraph, and shown in Fig. 3 on page D19-14 allows the pre-

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	**Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979**
	compressed intake air at pressure "P_a" shown in Figs. 6 and 8 on page D19-16 to flow between the combustion chamber and the intake manifold. See the Hiltner Declaration at paragraphs 11-21.
substantially during a majority portion of a compression stroke of the piston; and	Fig. 6, third diagram, illustrates intake valve closing at location 1 on the compression stroke after a majority of the compression stroke has occurred, with an intake air charging pressure of 10 bar. This comports with intake valve closing at greater than 90° after bottom dead center ("BDC") as shown in the lower portion of Fig. 8, curve α. See Hiltner Declaration at paragraphs 11-21. Additional discussion in Zappa *et al.* discloses "very delayed closing" and "retarded closing" of the "intake valves" at page 19-7, final two paragraphs.
operably controlling a fuel supply system to inject fuel into the combustion chamber after the intake valve is closed.	Page D19-10 describes fuel injection as fuel jets coming from the increased fuel injectors." Fig. 6, third diagram, on page D19-16 discloses conventional diesel combustion involving both a constant volume phase (illustrated by line segment 2-3) and a constant pressure phase (illustrated by line segment 3-4), which requires direct fuel injection at the end portion of the compression stroke and at the beginning of the expansion stroke. See the Hiltner Declaration at paragraphs 22-25.
2. The method of claim 1, wherein said selectively operating includes operating a variable intake valve closing mechanism to keep the intake valve open.	Fig. 3, page D19-14 shows an "inlet valve control device" that varies the timing of the closing of the intake valves by "delaying … the lifting phase" to keep the intake valves open. See page D19-3, last four paragraphs.
3. The method of claim 2, wherein the variable intake valve closing mechanism is operated at least one of hydraulically, pneumatically,	The "inlet valve control device" of Fig. 3 (page D19-14) of Zappa et al. is operated mechanically, as described on page D19-3, last four paragraphs.

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	**Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979**
mechanically, and electronically.	
4. The method of claim 1, wherein the selective operation of the air intake valve is based on at least one engine condition.	Zappa et al. discloses that "compression ratio [is] chosen in order to attain the same compression pressure," indicating that selective operation of the air intake valve is based on at least the engine condition of "compression pressure." See page D19-4, third paragraph. Page D19-3, fifth paragraph, indicates that "variation of the compression ratio is made virtually by changing the timing of the intake valves," and thus the intake valves operation is based on the engine condition of "compression pressure." Further, at page D19-7 Zappa et al. refers "varying the valve timing . . . during low speed operation," which discloses that selective operation of the air intake valve is based on the engine condition of engine speed.
5. The method of claim 1, wherein said selectively operating includes operating the intake valve to remain open for a portion of a second half of the compression stroke of the piston.	Figs. 6 and 8 of Zappa et al. (e.g., the diagram to the far right in Fig. 6) illustrate intake valve closing at 1 on the compression stroke after the beginning of the second half of the compression stroke. See the Hiltner Declaration at paragraphs 11-21.
6. The method of claim 1, wherein said operably controlling a fuel supply system includes operating a fuel injector assembly at least one of hydraulically, mechanically, and electronically.	Zappa et al. discusses a fuel system including fuel jets coming from fuel injectors to inject fuel into the combustion chamber. See page D19-10, final paragraph.
7. A method of operating an internal combustion engine	Zappa *et al.* discusses a "4-Stroke High Speed Diesel Engine," which is an internal combustion engine. *See* page D19-1, title. Page D19-2 provides a general description of the internal combustion engine, and pages D19-13 and 14 provide schematics and a photo of the internal

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979
	combustion engine.
including at least one cylinder and a piston slidable in the cylinder, the method comprising:	The "geometrical characteristics" described on page D19-2 include a "cylinder" with specified dimensions, and "pistons" are included as "engine components" in the last paragraph on that page. Figs. 15 and 16 on pages D19-20 and 21 illustrate the pistons and cylinders.
imparting rotational movement to a first turbine and a first compressor of a first turbocharger with exhaust air flowing from an exhaust port of the cylinder; imparting rotational movement to a second turbine and a second compressor of a second turbocharger with exhaust air flowing from an exhaust duct of the first turbocharger; compressing air drawn from atmosphere with the second compressor; compressing air received from the second compressor with the first compressor; supplying pressurized air from the first compressor to an air intake port of a combustion chamber in the cylinder via an intake manifold;	The engine described in Zappa *et al.* includes a "Two-stage Supercharging System" to pressurize air. *See* page D19-3, first - third paragraphs. The supercharging system "includes two turbochargers," each of which has a compressor and a turbine supplying pressurized air to air intake ports of combustion chambers in the engine cylinders via an intake manifold, as shown on page D19-13, Fig. 1. The first turbocharger is driven by the exhaust gases (see p. D19-4, penultimate paragraph, and page D19-8, second paragraph). See Hiltner Declaration at paragraphs 7 and 19.
controllably operating a fuel supply system to inject fuel directly into the combustion chamber; and	Page D19-10, final paragraph, describes fuel injection as "fuel jets coming from the increased fuel injectors."
selectively operating an air intake valve to open the air intake port to allow pressurized air	The "intake valves" described at the top of page D19-6 and illustrated in Figs. 15 and 16 on pages D19-20 and 21 are valves for the air intake

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979
to flow between the combustion chamber and the intake manifold	ports into the "combustion chamber" of the cylinders. Control of the "intake valves" as described on pages D19-3, ¶5, D19-7, penultimate paragraph, and shown in Fig. 3 on page D19-14 allows the pre-compressed intake air at pressure "P_a" shown in Figs. 6 and 8 on page D19-16 to flow between the combustion chamber and the intake manifold. See the Hiltner Declaration at paragraphs 11-21.
during a portion of a compression stroke of the piston,	Fig. 6, third diagram, illustrates intake valve closing at location 1 on the compression stroke after a majority of the compression stroke has occurred, with an intake air charging pressure of 10 bar. This comports with intake valve closing at greater than 90° after bottom dead center ("BDC") as shown in the lower portion of Fig. 8, curve α. See Hiltner Declaration at paragraphs 11-21. Additional discussion in Zappa *et al.* discloses "very delayed closing" and "retarded closing" of the "intake valves" at page 19-7, final two paragraphs.
wherein fuel is injected during a combustion stroke.	Fig. 6, third diagram, on page D19-16 discloses conventional diesel combustion involving a constant pressure phase (illustrated by line segment 3-4), which requires direct fuel injection at the beginning of the combustion stroke (i.e., expansion stroke). See Hiltner Declaration at paragraphs 22, 24, and 25. See also page D19-10, final paragraph, describing fuel jets injected into the combustion chamber.
8. The method of claim 7, wherein fuel injection begins during the compression stroke.	Zappa et al. discloses in the far right diagram of Fig. 6, a line segment 2-3 demonstrating an increase of pressure from 2 to 3. This pressure increase indicates fuel was injected prior to 2, during the compression stroke, after the intake valve was closed at 1. See the Hiltner Declaration at paragraphs 22, 23, and 25.
9. The method of claim 7, wherein said	Zappa et al. discloses an "inlet valve control device" that varies the

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979
selectively operating includes operating a variable intake valve closing mechanism to interrupt cyclical movement of the intake valve.	timing of the closing of the intake valves. See page D19-3, last four paragraphs, and page D19-14, Fig. 3. Page D19-8 describes how the "timing control device for the intake valve … performs … a shifting of the whole intake phase," and thus interrupts the cyclical movement of the intake valve during the phase shift.
10. The method of claim 7, wherein the selective operation of the air intake valve is based on at least one engine condition.	Zappa et al. discloses that "compression ratio [is] chosen in order to attain the same compression pressure," indicating that selective operation of the air intake valve is based on at least the engine condition of "compression pressure." See page D19-4, third paragraph. Note also page D19-3, fifth paragraph, indicating compression ratio is changed by the timing of the intake valve. Further, at page D19-7 Zappa et al. refers "varying the valve timing . . . during low speed operation," which discloses that selective operation of the air intake valve is based on the engine condition of engine speed
11. The method of claim 7, wherein said selectively operating includes operating the intake valve to remain open for a portion of a second half of the compression stroke of the piston.	Figs. 6 and 8 of Zappa et al. (e.g., the diagram to the far right in Fig. 6) illustrate intake valve closing at 1 on the compression stroke after the beginning of the second half of the compression stroke. See the Hiltner Declaration at paragraphs 11-21.
12. The method of claim 7, wherein said controllably operating a fuel supply system includes operating a fuel injector assembly at least one of hydraulically, mechanically, and electronically.	Page D19-10 describes fuel injection as "fuel jets coming from the increased fuel injectors."
13. An internal combustion engine, comprising:	Zappa *et al.* discusses a "4-Stroke High Speed Diesel Engine," which is an internal combustion engine. *See* page D19-1, title. Page D19-2

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979
	provides a general description of the internal combustion engine, and pages D19-13 and 14 provide schematics and a photo of the internal combustion engine.
a block defining at least one cylinder;	Page D19-13, Fig. 1 shows the engine with an engine block and multiple cylinders. See page D19-13, Fig. 1.
a head connected with said block, said head having an air intake port and an exhaust port;	Zappa et al. discloses "cyl. heads" and "cylinder head" at page D19-2, final paragraph, and page D19-10, second-to-last paragraph, respectively. Zappa also illustrates a cylinder head with intake and exhaust ports at page D19-20, Fig. 15a and page D19-21, Fig. 16, upper left view.
a piston slidable in each cylinder;	Zappa et al. discloses engine cylinders and pistons at page D19-2. See also page D19-20, Fig. 15, and page D19-21, Fig. 16, illustrating pistons and cylinders.
an air intake valve controllably movable to open and close the air intake port;	Zappa et al. discloses intake valves and intake ports at page D19-20, Fig. 15 and page D19-21, Fig. 16. In addition, Zappa discusses intake valve control on page D19-3, fifth paragraph, and page D19-7, penultimate paragraph, with an illustration of an "intake valve control device" at page D19-14, Fig. 3. See also, Hiltner Declaration at paragraphs 11-21.
a first turbocharger including a first turbine coupled with a first compressor, the first turbine being in fluid communication with the exhaust port, the first compressor being in fluid communication with the air intake port; a second compressor being in fluid	Zappa et al. discloses a two-stage supercharging system including series turbochargers having turbines and compressors supplying pressurized air to an intake port of a combustion chamber in the engine cylinders via an intake manifold. See page D19-3, first - third paragraphs, and page D19-13, Fig. 1. See also, the Hiltner Declaration at paragraphs 7 and 19.

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979
communication with atmosphere and the first compressor;	
a fuel supply system operable to controllably inject fuel into the combustion chamber; and	Page D19-10 describes fuel injection as "fuel jets coming from the increased fuel injectors."
a controller configured to selectively operate the air intake valve to remain open	Zappa et al. discloses controlling the intake valves with an "intake valve control device" at page D19-3, fifth paragraph and page D19-7, penultimate paragraph, and page D19-14, Fig. 3.
during a portion of a compression stroke of the piston,	Figs. 6 and 8 of Zappa et al. (e.g., the diagram to the far right in Fig. 6) illustrate intake valve closing at 1 on the compression stroke after the beginning of the second half of the compression stroke. See the Hiltner Declaration at paragraphs 11-21.
wherein the controller is configured to inject fuel into the combustion chamber during an combustion stroke.	Fig. 6, third diagram, on page D19-16 discloses conventional diesel combustion involving a constant pressure phase (illustrated by line segment 3-4), which requires direct fuel injection at the beginning of the combustion stroke (i.e., expansion stroke). See Hiltner Declaration at paragraphs 22, 24, and 25. See also page D19-10, final paragraph, describing fuel jets injected into the combustion chamber.
14. The engine of claim 13, wherein said second compressor is coupled with said first turbine.	Zappa et al. discloses a two-stage supercharging system including series turbochargers having turbines and compressors. See page D19-3, first - third paragraphs, and page D19-13, Fig. 1.

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	**Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979**
15. The engine of claim 13, wherein the controller is configured to operate the intake valve to remain open for a portion of a second half of the compression stroke of the piston.	Figs. 6 and 8 of Zappa et al. (e.g., the diagram to the far right in Fig. 6) illustrate intake valve closing at 1 on the compression stroke after the beginning of the second half of the compression stroke. See the Hiltner Declaration at paragraphs 11-21.
16. The engine of claim 13, wherein the fuel supply system includes a fuel injector assembly.	Page D19-10 describes fuel injection as "fuel jets coming from the increased fuel injectors."
21. A method of controlling an internal combustion engine having a variable compression ratio,	Zappa *et al.* discusses a "4-Stroke High Speed Diesel Engine," which is an internal combustion engine. *See* page D19-1, title. Page D19-2 provides a general description of the internal combustion engine, and pages D19-13 and 14 provide schematics and a photo of the internal combustion engine. Zappa et al. also discloses the engine operating with a variable compression ratio. See page D19-2, lines 22-25 and page D19-3, fifth - eighth paragraphs.
said engine having a block defining a cylinder,	Zappa et al. illustrates the engine with an engine block and multiple cylinders. See page D19-13, Fig. 1.
a piston slidable in said cylinder,	Zappa et al. discloses engine cylinders and pistons at page D19-2. See also page D19-20, Fig. 15, and page D19-21, Fig. 16, illustrating pistons and cylinders.

COMPARISON OF ’280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	**Exemplary Disclosure in Zappa, et al., “A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio,” CIMAC, 1979**
a head connected with said block, said piston, said cylinder, and said head defining a combustion chamber, the method comprising:	Zappa et al. discloses “cyl. heads” and “cylinder head” at page D19-2, final paragraph, and page D19-10, second-to-last paragraph, respectively. Zappa et al. also illustrates a cylinder head with intake and exhaust ports at page D19-20, Fig. 15a and page D19-21, Fig. 16, upper left view. The “combustion chamber” is described on page D19-4, ¶1, page D19-5, ¶3, and D19-8, ¶4.
pressurizing air; supplying said air to an intake manifold of the engine;	The engine described in Zappa *et al.* includes a “Two-stage Supercharging System” to pressurize air. *See* page D19-3. Fig. 1 on page D19-13 shows an intake manifold after the “two-stage supercharging system,” and page D19-6 mentions a “pressure drop between air and gas manifold pressure.” The “intake valves” described on pages D19-3, ¶3, D19-6, ¶1, D19-7, penultimate paragraph, and illustrated in Figs. 15 and 16 on pages D19-20 and 21 are valves for the air intake ports into the combustion chambers of the cylinders. Figs. 6 and 8 on pages D19-16 show the “supercharging pressure” (“P_a”) of the pressurized air supplied to the intake port.
maintaining fluid communication between said combustion chamber and the intake manifold during a portion of an intake stroke	Zappa et al. discloses an intake valve being open through a portion of the compression stroke. See far right diagram of Fig. 6 on page D19-16. See also the Hiltner Declaration at paragraphs 11-21.
and through a predetermined portion of a compression stroke; and	Zappa *et al.* discloses “very delayed closing” and “retarded closing” of the “intake valves” at page 19-7, final two paragraphs. Fig. 6, third diagram, illustrates intake valve closing at location 1 on the compression stroke after a majority of the compression stroke has occurred, with an intake air charging pressure of 10 bar. This comports with intake valve closing at greater than 90° after bottom dead center (“BDC”) as shown in the lower portion of Fig. 8, curve α. See the Hiltner Declaration at paragraphs 11-21.

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979
supplying a pressurized fuel directly to the combustion chamber during a portion of a combustion stroke.	Fig. 6, third diagram, on page D19-16 discloses conventional diesel combustion involving a constant pressure phase (illustrated by line segment 3-4), which requires direct fuel injection at the beginning of the combustion stroke (i.e., expansion stroke). See Hiltner Declaration at paragraphs 22, 24, and 25. See also page D19-10, final paragraph, describing fuel jets injected into the combustion chamber.
22. The method of claim 21, further including supplying the pressurized fuel during a portion of the compression stroke.	Zappa et al. discloses in the far right diagram of Fig. 6, a line segment 2-3 demonstrating an increase of pressure from 2 to 3. This pressure increase indicates fuel was injected prior to 2, during the compression stroke, after the intake valve was closed at 1. See the Hiltner Declaration at paragraphs 22, 23, and 25.
23. The method of claim 22, wherein supplying the pressurized fuel includes supplying a pilot injection at a predetermined crank angle before a main injection.	Zappa et al. discusses a fuel system including fuel jets coming from fuel injectors to inject fuel into the combustion chamber. See page D19-10, final paragraph.
24. The method of claim 23, wherein said main injection begins during the compression stroke.	Zappa et al. discloses in the far right diagram of Fig. 6, a line segment 2-3 demonstrating an increase of pressure from 2 to 3. This pressure increase indicates fuel was injected prior to 2, during the compression stroke, after the intake valve was closed at 1. See the Hiltner Declaration at paragraphs 22, 23, and 25.

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979
25. The method of claim 21, wherein said predetermined portion of the compression stroke is at least a majority of the compression stroke.	Fig. 6, third diagram, illustrates intake valve closing at location 1 on the compression stroke after a majority of the compression stroke has occurred, with an intake air charging pressure of 10 bar. This comports with intake valve closing at greater than 90° after bottom dead center ("BDC") as shown in the lower portion of Fig. 8, curve α. See the Hiltner Declaration at paragraphs 11-21.
26. The method of claim 21, wherein said pressurizing includes a first stage of pressurization and a second stage of pressurization.	Zappa et al. discloses a two-stage supercharging system including series turbochargers having turbines and compressors supplying pressurized air to an intake port of a combustion chamber in the engine cylinders via an intake manifold. See page D19-3, first - third paragraphs, and page D19-13, Fig. 1. See also, the Hiltner Declaration at paragraphs 7 and 19.
27. The method of claim 26, further including cooling air between said first stage of pressurization and said second stage of pressurization.	Zappa et al. discloses a "low pressure cooler" downstream of one turbocharger and upstream of another turbocharger. See page D19-3, first - third paragraphs, and page D19-13, Fig. 1.
28. The method of claim 21, further including cooling the pressurized air.	Zappa et al. discloses a "low pressure cooler" downstream of one turbocharger and upstream of another turbocharger. See page D19-3, first - third paragraphs, and page D19-13, Fig. 1.

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	**Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979**
29. A method of controlling an internal combustion engine having a variable compression ratio,	Zappa *et al.* discusses a "4-Stroke High Speed Diesel Engine," which is an internal combustion engine. *See* page D19-1, title. Page D19-2 provides a general description of the internal combustion engine, and pages D19-13 and 14 provide schematics and a photo of the internal combustion engine. Zappa et al. also discloses the engine operating with a variable compression ratio. See page D19-2, lines 22-25 and page D19-3, fifth - eighth paragraphs.
said engine having a block defining a cylinder,	Zappa et al. illustrates the engine with an engine block and multiple cylinders. See page D19-13, Fig. 1.
a piston slidable in said cylinder,	Zappa et al. discloses engine cylinders and pistons at page D19-2. See also page D19-20, Fig. 15, and page D19-21, Fig. 16, illustrating pistons and cylinders.
a head connected with said block, said piston, said cylinder, and said head defining a combustion chamber, the method comprising:	Zappa et al. discloses "cyl. heads" and "cylinder head" at page D19-2, final paragraph, and page D19-10, second-to-last paragraph, respectively. Zappa also illustrates a cylinder head with intake and exhaust ports at page D19-20, Fig. 15a and page D19-21, Fig. 16, upper left view. The "combustion chamber" is described on page D19-4, ¶1, page D19-5, ¶3, and D19-8, ¶4.
pressurizing air to a ratio of at least 4:1 with respect to atmospheric pressure;	Zappa et al. discloses a "Two-stage Supercharging System" pressurizing intake air to 10 bar. In particular, the far right diagram of Fig. 6, show the "supercharging pressure" ("P_a") of the pressurized air supplied to the intake port. See also page D19-3 and Fig. 7. In addition, Fig. 8 discloses a charging air pressure horizontal scale extending numerically up to 9 bar.

COMPARISON OF '280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., "A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio," CIMAC, 1979
supplying the pressurized air to an intake manifold of the engine;	Zappa et al. discloses in Fig. 1 of page D19-13 an intake manifold after the "two-stage supercharging system," and page D19-6 mentions a "pressure drop between air and gas manifold pressure." The "intake valves" described on pages D19-3, ¶3, D19-6, ¶1, D19-7, penultimate paragraph, and illustrated in Figs. 15 and 16 on pages D19-20 and 21 are valves for the air intake ports into the combustion chambers of the cylinders.
maintaining fluid communication between the combustion chamber and the intake manifold during an intake stroke	Zappa et al. discloses an intake valve being open through a portion of the compression stroke. See far right diagram of Fig. 6 on page D19-16. See also the Hiltner Declaration at paragraphs 11-21.
and a majority of a compression stroke; and	Zappa et al. in the far right diagram of Fig. 6 illustrates intake valve closing at location 1 on the compression stroke after a majority of the compression stroke has occurred. This comports with intake valve closing at greater than 90° after bottom dead center ("BDC") as shown in the lower portion of Fig. 8, curve α. See the Hiltner Declaration at paragraphs 11-21. Zappa *et al.* also discloses "very delayed closing" and "retarded closing" of the "intake valves" at page 19-7, final two paragraphs.
supplying a fuel to the combustion chamber during at least a portion of the remaining compression stroke.	Zappa et al. discloses in the far right diagram of Fig. 6, a line segment 2-3 demonstrating an increase of pressure from 2 to 3. This pressure increase indicates fuel was injected prior to 2, during the compression stroke, after the intake valve was closed at 1. Conventional diesel combustion involves a constant volume phase (illustrated by line segment 2-3), which requires direct fuel injection at the end portion of the compression stroke. See the Hiltner Declaration at paragraphs 22, 23, and 25.

COMPARISON OF ’280 PATENT CLAIMS TO ZAPPA ET AL.

Claims of U.S. Patent No. 6,688,280	Exemplary Disclosure in Zappa, et al., “A 4-Stroke High Speed Diesel Engine With Two-Stage Of Supercharging And Variable Compression Ratio,” CIMAC, 1979
30. The method of claim 29, wherein said majority is at least 90 degrees crank angle after bottom dead center.	Zappa et al. in the far right diagram of Fig. 6 illustrates intake valve closing at location 1 on the compression stroke after a majority of the compression stroke has occurred. This comports with intake valve closing at greater than 90° after bottom dead center (“BDC”) as shown in the lower portion of Fig. 8, curve α. See the Hiltner Declaration at paragraphs 11-21.
31. The method of claim 29, wherein said supplying fuel includes injecting a first portion of fuel a predetermined period prior to injecting a second portion of fuel.	Zappa et al. discusses a fuel system including fuel jets coming from fuel injectors to inject fuel into the combustion chamber. See page D19-10, final paragraph.
32. The method of claim 31, wherein said injecting the second portion of fuel begins during the compression stroke and terminates during a combustion stroke.	Zappa et al. discloses in the far right diagram of Fig. 6, a line segment 2-3 demonstrating an increase of pressure from 2 to 3. This pressure increase indicates fuel was injected prior to 2, during the compression stroke, after the intake valve was closed at 1. Zappa et al. also discloses in the far right diagram of Fig. 6 a line segment 3-4 demonstrating an increase in volume from 3-4 with no decrease in pressure, indicating fuel was injected during the combustion stroke (i.e., expansion stroke). See Hiltner Declaration at paragraphs 22-25. See also page D19-10, final paragraph, describing fuel jets injected into the combustion chamber.
33. The method of claim 31, further including cooling the air prior to supplying the air to the combustion chamber.	Zappa et al. discloses a “low pressure cooler” downstream of one turbocharger and upstream of another turbocharger. See page D19-3, first - third paragraphs, and page D19-13, Fig. 1.

PATENT
Customer No. 22,852
Attorney Docket No. 08351.0294

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

In re *Ex Parte* Reexamination of U.S. Patent 6,688,280	Group Art Unit: Unassigned
Reexamination Control No: Unassigned	Examiner: Unassigned
Filed: January 20, 2006	
For: AIR AND FUEL SUPPLY SYSTEM FOR COMBUSTION ENGINE	

Mail Stop *Ex Parte* Reexam
Hon. Commissioner for Patents
P.O. Box 1450
Alexandria, VA 22313-1450

Sir:

Declaration of Dr. Joel Hiltner

I, Dr. Joel Hiltner, declare as follows:

1. I have a great deal of experience and familiarity with the art of internal combustion engines. I received a Ph.D. in Mechanical Engineering, a Master's in Mechanical Engineering, and a Bachelor of Science in Mechanical Engineering (Summa Cum Laude), all from The Ohio State University in 1997, 1993, and 1992, respectively. The research topic for my Master's and Ph.D. programs involved optical diagnostics in internal combustion engines. After receiving my doctorate, I was a post-doctoral research fellow at Trinity College in Dublin, Ireland; a visiting research fellow at the University of California at Berkley; and a visiting fellow at RMIT University in Melbourne, Australia. During my work at the University of California, I assisted in developing an

HCCI type internal combustion engine aimed at high efficiency and low NO_x operation. At RMIT University, I developed a senior level class covering automotive engine and transmission design.

2. Currently, I am the chief engineer for Hiltner Combustion Systems. My work for the past six years has focused on efficiency improvement and emission reduction through combustion system improvement for all varieties of combustion engines. My efforts at Hiltner Combustion Systems have included modeling of non-traditional engines and engine cycles, optimization of spark ignited natural gas engines for stationary power generation, and the development of experimental and analytical tools for engine test cell applications. I have conducted work pertaining to the thermodynamics, combustion chemistry, and control systems of both diesel and spark ignited internal combustion engines. I have also collaborated with a number of universities and researchers on advanced research and development topics focused on low emission combustion technologies. In addition, I have actively participated in several technical organizations related to engines and published several papers. My curriculum vitae is attached as Exhibit A.

3. In my role at Hiltner Combustion Systems, I have assisted Caterpillar Inc., the owner of U.S. Patent No. 6,688,280 ("the '280 patent"), and I have also provided assistance to MaK and Perkins Engine Company, which are both subsidiaries of Caterpillar Inc. I am named as an inventor on at least seven U.S. patents owned by Caterpillar Inc.

4. I am familiar with the level of ordinary skill in the art of internal combustion engines as of May 14, 2002.

5. I have reviewed the '280 patent. I have also reviewed the following publication: Zappa et al., "A 4-STROKE HIGH SPEED DIESEL ENGINE WITH TWO-STAGE OF SUPERCHARGING AND VARIABLE COMPRESSION RATIO;" CIMAC, 1979 ("Zappa et al.").

ZAPPA ET AL. IS MUCH MORE RELEVANT TO THE '280 PATENT THAN THE BRYANT DOCUMENT AT ISSUE IN THE *INTER PARTES* REEXAMINATION OF THE '280 PATENT

6. Caterpillar Inc. filed two of my prior declarations in the *inter partes* reexamination proceeding for the '280 patent (Control No. 95/000,050). While those declarations address several issues, a primary area of focus concerns PCT International Publication No. WO 98/02653 to Bryant ("Bryant"), which is the primary reference applied in claim rejections in that proceeding. For example, my previous declarations explain several deficiencies in Bryant, especially Bryant's scattered discussion setting forth numerous listings of alternative features without providing any meaningful disclosure of how to combine particular engine features, such as any particular intake valve timing, fuel supply, air compression/non-compression, etc. My prior declarations also address Bryant's lack of disclosure of turbochargers, as well as Bryant's failure to disclose variable compression ratio.

7. One of ordinary skill in the art would understand that Zappa et al. is much more relevant to the '280 patent than the Bryant document at issue in the *inter partes* reexamination of the '280 patent. For example, in contrast to Bryant's catalog of unconnected discussions, Zappa et al. discloses several features that are directly linked to one another. In particular, as discussed in more detail below, Zappa et al. provides specific disclosure of engines and methods of engine operation that include both an

intake valve closing extremely late in the compression stroke (e.g., after a majority portion of the compression stroke) and fuel being injected directly into a combustion chamber during both the compression and expansion strokes, after the intake valve closes (in addition to also disclosing an intake valve closing extremely early in the intake stroke in combination with direct fuel injection). As discussed in my prior declarations, no such disclosure of late intake valve closing in combination with fuel injection timing is contained in Bryant. Also, rather than having no mention of turbochargers and variable compression ratio, Zappa et al. discloses both series turbochargers (e.g., at page D19-3, first - third paragraphs, and page D19-13, Fig. 1) and variable compression ratio (e.g., at page D19-2, lines 22-25, referring to "main features" including "variable compression ratio," and page D19-3, fifth - eighth paragraphs describing "Variable Compression Ratio").

FIG. 6 of ZAPPA ET AL.

8. As reflected by its title, Zappa et al. discloses a four-stroke, diesel cycle engine. By definition, a four-stroke, diesel engine involves a piston sliding in a cylinder in the following successive strokes: an intake stroke, a compression stroke, an expansion stroke, and an exhaust stroke. In the intake and expansion strokes, the piston starts at a top dead center (TDC) location in the cylinder and moves downward in the cylinder to a bottom dead center (BDC) location in the cylinder. In the compression and exhaust strokes, the piston moves upward in the cylinder from the BDC location to the TDC location. Cylinder volume above the piston varies while the piston moves in each of the strokes, with the volume being at a maximum when the piston is at its BDC location, and at a minimum when the piston is at its TDC location. Pressure in the

cylinder during piston movement depends on whether intake and exhaust valves associated with the cylinder are in an open or closed position and on the combustion conditions in the cylinder. When the intake and exhaust valves are closed, movement of the piston upward toward the TDC location causes an increase in cylinder pressure, and movement of the piston downward toward the BDC location causes a reduction in cylinder pressure. Opening intake and/or exhaust valves places the cylinder in flow communication with the respective intake/exhaust manifold causing the pressure of the cylinder to reach equilibrium with pressure of the intake/exhaust manifold. Combustion in the cylinder causes an increase in pressure within the cylinder.

9. Fig. 6 on page D19-16 of Zappa et al. includes three different diagrams illustrating operation of the four-stroke, diesel-cycle engine disclosed in Zappa et al. The diagram on the far right of Fig. 6 illustrates engine operation involving intake air pre-compressed to a charge air pressure (P_a) of 10 bar; the diagram in the middle shows engine operation with a P_a of 5 bar; and the diagram on the far left of Fig. 6 shows an engine operating with a P_a of 2.9 bar pressure. Each diagram is in a graphical form having cylinder volume (V) extending along the X-axis and cylinder pressure (P) extending along the Y-axis. In each diagram, the line 1-2-3-4-5 represents pressure in the cylinder and volume above the piston while the piston moves through its strokes. Location 2 on the line represents TDC in the compression stroke (which is TDC in the expansion stroke) and location 5 on the line represents BDC in the expansion stroke (which is BDC in the exhaust stroke). Segment 2-3-4-5 of the line represents the expansion stroke.

10. In Fig. 6 of Zappa et al., location 1 identifies when compression begins in compression stroke. In the far left diagram of Fig. 6, location 1 appears to be located at the beginning of the compression stroke (BDC), thus indicating that the intake valve(s) was/were closed when the piston reached BDC of the intake stroke so that the upward movement of the piston causes increasing cylinder pressure throughout the entire compression stroke. In both the middle and far right diagrams of Fig. 6, the location 1 is offset from BDC of the compression stroke and located in-between BDC and TDC of the compression stroke. This corresponds to engine operation where the intake valve(s) was/were closed late during the compression stroke or engine operation where the intake valve(s) was/were closed early during the intake stroke, resulting in the upward piston movement only causing a pressure increase after the piston has traveled a distance beyond BDC of the compression stroke. This is commonly referred to as the Miller cycle.

ZAPPA ET AL. DISCLOSES CLOSING AN INTAKE VALVE MORE THAN 90 CRANK ANGLE DEGREES AFTER BOTTOM DEAD CENTER OF A PISTON'S COMPRESSION STROKE AND LESS THAN 90 CRANK ANGLE DEGREES AFTER TOP DEAD CENTER OF A PISTON'S INTAKE STROKE

11. As explained in the following paragraphs, one of ordinary skill in the art would understand that in Zappa et al. at page D19-16, Figs. 6 and 8 show piston-cylinder compression beginning later than half way through the compression stroke. Zappa et al. points out that this is achieved through late intake valve closing or early intake valve closing. (See, e.g., the sentence bridging pages D19-5 and D19-6, which refers to both "anticipated" (i.e., early) and "delayed" (i.e., late) intake valve closing in a discussion of Miller cycle engine operation preceding a discussion of Fig. 6.)

Disclosure in Fig. 6 regarding very late/early intake valve closing

12. Attached Exhibit B is an annotated version of the diagram shown on the far right of Fig. 6 of Zappa et al. Along the X-axis, the point "TDC" corresponds to the cylinder volume above the piston at top dead center, and the point "BDC" corresponds to cylinder volume above the piston at bottom dead center. The line 1-2-3-4-5 schematically represents pressure in the cylinder and cylinder volume above the piston during a 4-stroke diesel cycle of the engine.

13. As mentioned above, the location 1 of the line 1-2-3-4-5 shows the beginning of an in-cylinder pressure increase caused by upward piston movement. Exhibit B shows that the distance along the X-axis from the location 1 to the piston TDC volume is shorter than the distance along the X-axis to the piston BDC volume. (In Exhibit B, this is illustrated by the X-axis position labeled "113 degrees after BDC" being closer to TDC than BDC.) This relative position of location 1 discloses that piston compression begins after the piston passes through more than half of the compression stroke (i.e., greater than 90 crank angle degrees after BDC of the compression stroke). The fact that compression does not begin until more than half way through the compression stroke, indicates that either the intake valve(s) was/were held open past the half way point of the compression stroke, or the intake valve(s) was/were closed well before the midpoint of the intake stroke. The text of Zappa et al. points out that either late (i.e., "delayed" or "retarded") intake valve closing or early (i.e., "anticipated") intake valve closing can be used to achieve Miller cycle engine operation (e.g., in the sentence bridging pages D19-5 - D19-6 and at page D19-7, final two paragraphs). Accordingly, the far right diagram

of Fig. 6 discloses very late intake valve closing and it also discloses very early intake valve closing.

14. The relative spacing of location 1 closer to the TDC volume than the BDC volume in Exhibit B discloses an intake valve being held open through more than 50% of a compression stroke (i.e., through a majority portion of the compression stroke), and it also discloses an alternative arrangement having an intake valve being held open for only less than 50% of an intake stroke (i.e., through a minority of the intake stroke). Stated another way, the position of location 1 discloses an intake valve being closed more than 90 crank angle degrees after bottom dead center of the compression stroke, and it also discloses an alternative arrangement where an intake vale is closed less than 90 crank angle degrees after top dead center of the intake stroke.

15. The three diagrams shown in Fig. 6 of Zappa et al. provide concrete examples of engine operating conditions. In each case, in-cylinder compression begins at a different place in the compression stroke (location 1 in the diagrams) providing a different effective compression ratio. The value of the effective compression ratio is identified in each of the three diagrams by using the symbol "ρ." For the diagram on the far left of Fig. 6, compression begins at bottom dead center of the compression stroke and, thus, the effective compression ratio (which is identified in Fig. 6 as being 11.2) is equal to the geometric compression ratio, which is defined as follows:

$$\text{Geometric Compression Ratio} = \frac{\text{cylinder volume at bottom dead center}}{\text{cylinder volume at top dead center}}$$

The effective compression ratios for the other two cases (middle and far right diagrams of Fig. 6) have smaller values of 7.6 and 4.6, respectively. This show a progressively

later start of compression, indicating progressive movement of the intake valve closing location away from bottom dead center.

16. Although Zappa et al. does not explicitly state the crank angle corresponding to the beginning of compression (location 1) for each of the diagrams of Fig. 6, one of ordinary skill in the art would be able to calculate this information given the engine details provided in Zappa et al. and equations that are well known to those of ordinary skill in the art. By using those details disclosed in Zappa et al. and well-known equations, I prepared Exhibit C, which shows the relationship between 1) the crank angle at which in-cylinder compression begins, and 2) the effective compression ratio, for the engine operations of Fig. 6 of Zappa et al. As illustrated in Exhibit C, for the lowest effective compression ratio (4.6) associated with the far right diagram of Fig. 6, in-cylinder compression begins at 113 crank angle degrees after bottom dead center of the compression stroke, which is well after the half-way location of the compression stroke. This confirms that the far right diagram of Fig. 6 discloses location 1 being 113 crank angle degrees after bottom dead center of the compression stroke. As mentioned above, Zappa et al. discloses that both late and early intake valve closings are used to achieve a late start of compression (e.g., at page D19-7). Accordingly, one of ordinary skill in the art would understand that the far right diagram of Fig. 6 shows both an intake valve closing very late at 113 crank angle degrees after bottom dead center of the compression stroke, and an intake valve closing very early at 113 crank angle degrees before bottom dead center of the intake stroke.

17. To arrive at the results shown in Exhibit C, the cylinder volume at the start of compression was calculated from the effective compression ratios provided by Zappa et

al. (7.6 for the middle diagram of Fig. 6 and 4.6 for the far right hand diagram of Fig. 6). From the effective compression ratio (and the bore and stroke provided in Zappa et al.), the cylinder volume at the start of compression (the position of location 1) is determined from the following relationship:

$$\textit{Effective Compression Ratio} = \frac{\textit{cylinder volume at start of compression}}{\textit{cylinder volume at top dead center}}.$$

The volume at top dead center for the engine described in Zappa et al. is 1.099 liters, which, when combined with the equation above, gives start of compression volumes of 8.35 liters (for the effective compression ratio of 7.6) and 5.05 liters (for the effective compression ratio of 4.6). These start of compression volumes can then be used to determine the crank angle at which compression begins by making a standard assumption of connecting rod length and using the following well-known equation that enables determination of crank angle (where "cad" is the crank angle and "R" is the ratio between the connecting rod length and the crank radius):

Cylinder Volume = Volume at TDC + Volume Displaced / 2 * (R + 1 - COS(cad) - SQRT(R ^ 2 - SIN(cad)^2))

Disclosure in Fig. 6 regarding precompressed intake air in combination with very late/early intake valve closing

18. One of ordinary skill in the art would understand that the far right diagram of Fig. 6 of Zappa et al. also shows engine operation involving intake of air precompressed to 10 bar before an intake valve is closed very late (i.e., more than 90 crank angle degrees after bottom dead center) in the compression stroke, and engine operation involving intake of air precompressed to 10 bar before an intake valve is closed very early (i.e., more than 90 crank angle degrees before bottom dead center) in the intake stroke. This is illustrated by the fact that the far right diagram of Fig. 6 refers to a

charge air pressure (P_a) of 10 bar, as well as showing that location 1 is spaced in a Y-axis direction above the X-axis. The horizontal line on which location 1 lies in Exhibit B is representative of the intake air being pre-compressed to 10 bar. (Note that in Fig. 6 of Zappa et al., the far right diagram has location 1 positioned further in the Y-axis direction than the location 1 shown in the middle and far left diagrams, which are associated with lower charge air pressures (P_a) of 5 bar and 2.9 bar, respectively.)

19. Zappa et al. (e.g., at page D19-3) describes an engine arrangement including a "Two-stage Supercharging System" involving a pair of turbochargers in series with one another, as well as a first intercooler positioned between the pair of turbochargers ("low pressure cooler") and a second intercooler located downstream of the turbocharger pair ("high pressure cooler"). Consequently, the intake air for the engine operation of Exhibit B would be precompressed by series turbochargers and cooled by intercoolers.

Disclosure in Fig. 8 regarding very late/early intake valve closing

20. Attached Exhibit D is an annotated version of the bottom portion of Fig. 8 of Zappa et al. Exhibit D includes a curve α representing the theoretical start of engine cylinder compression in terms of crank angle degrees after bottom dead center of the compression stroke. As reflected by the portion of curve α passing into the cross-hatched portion of Exhibit D, curve α shows cylinder compression starting more than 90 degrees after bottom dead center of the compression stroke. Therefore, Fig. 8 of Zappa et al. discloses an intake valve being closed more than 90 degrees after bottom dead center of the compression stroke and also discloses an intake valve being closed less than 90 degrees after top dead center of the intake stroke. Furthermore, Fig. 8 of

Zappa et al. shows such ultra late and ultra early intake valve closing as providing a reduction of effective compression ratio, as reflected by the downward-sloped curve labeled "ρ," which has its scale on the left side of Fig. 8.

Summary of disclosure of Figs. 6 and 8

21. One of ordinary skill in the art would understand that Figs. 6 and 8 disclose an air intake valve being selectively operated to allow pressurized air to flow between a combustion chamber and an intake manifold substantially during a majority portion of a compression stroke of a piston, as well as also during only the first half of the intake stroke. In addition, one of ordinary skill in the art would understand that Figs. 6 and 8 of Zappa et al. disclose fluid communication between a combustion chamber and an intake manifold being maintained during at least a portion of an intake stroke and through a majority of a compression stroke, as well as also disclosing fluid communication between the combustion chamber and the intake manifold being maintained during only a minority of the intake stroke. In particular, Figs. 6 and 8 of Zappa et al. disclose an intake valve being held open through an intake stroke and through more than 50% of a compression stroke, as well as also disclosing an intake valve being held open for only less than 50% of the intake stroke.

ZAPPA ET AL. DISCLOSES INJECTING FUEL DIRECTLY INTO A COMBUSTION CHAMBER AFTER AN INTAKE VALVE IS CLOSED AND DURING AT LEAST PORTIONS OF BOTH THE COMPRESSION AND EXPANSION STROKES

22. One of ordinary skill in the art would understand that in Zappa et al. at page D19-16, Fig. 6 discloses fuel being injected directly into a combustion chamber after an intake valve closes. Fig. 6 of Zappa et al. also discloses fuel being injected

directly into a combustion chamber during at least a portion of a compression stroke and during at least a portion of an expansion stroke.

23. Referring to Exhibit B, line segment 2-3 represents an increase in cylinder pressure without any substantial change in cylinder volume above a piston, and location 3 is at a pressure that Zappa et al. refers to as the maximum "combustion pressure." In light of the fact that Zappa et al. refers to direct injection of diesel fuel (for example, at page D19-10 in the last paragraph), one of ordinary skill in the art would understand that the increase in pressure illustrated by line segment 2-3 is caused by fuel being directly injected into the cylinder during the compression stroke after the intake valve closes at location 1 (and before the end of the compression stroke at location 2). The fuel injection leading to the combustion pressure spike of line segment 2-3 would need to be injected during the compression stroke after the intake valve closing of location 1 in order to avoid premature combustion of diesel fuel.

24. As shown in Exhibit B, line segment 3-4 represents a relatively constant "combustion pressure" in the cylinder being maintained while the cylinder volume above the piston increases due to the piston moving from top dead center in the expansion stroke. The far right diagram of Fig. 6 also shows an increase in temperature from 844 degrees C to 1712 degrees C during the constant combustion pressure of line segment 3-4. One of ordinary skill in the art would understand that line segment 3-4 and the temperature increase illustrate a constant pressure phase of combustion in the cylinder caused by fuel being injected into the combustion chamber after top dead center during the expansion stroke. Therefore, Exhibit B shows conventional diesel combustion involving both a constant volume phase (illustrated by line segment 2-3) and a constant

pressure phase (illustrated by line segment 3-4), which requires direct fuel injection at the end portion of the compression stroke and at the beginning of the expansion stroke.

25. Attached Exhibit E includes pages 142-144 and 153-154 of Obert, Internal Combustion Engines *Analysis and Practice*, second ed., 1950. In Exhibit E, at page 153, the P-V diagram of Fig. 6-4 illustrates the same compression-ignition (CI) engine cycle as the compression ignition engine cycle shown in the Fig. 6 of Zappa. In particular, Fig. 6-4 of Exhibit E shows a constant-volume, combustion pressure increase b-b' that is the same as the constant-volume, combustion pressure increase 2-3 of Fig. 6 of Zappa et al., and Fig. 6-4 of Exhibit E also shows a constant-pressure, combustion volume increase b'-c that is the same as the constant-pressure, combustion volume increase 3-4 of Fig. 6 of Zappa et al. Further, in the text associated with Fig. 6-4, Exhibit E at page 153 specifically refers to the pressure diagram of Fig. 5-12 (page 143), which clearly shows "injection start" and "injection 100" indicating that fuel injection is initiated before TDC of the compression stroke and then continued through the end of the compression stroke beyond TDC and into the beginning of the expansion stoke. As understood by one of ordinary skill in the art, that timing of fuel injection provides the combustion pressure-volume relationships illustrated in both Fig. 6-4 of Exhibit E and Fig. 6 of Zappa et al. Accordingly, the pages of Obert in Exhibit E confirm that the far right diagram of Fig. 6 of Zappa et al. discloses fuel being injected at the end portion of the compression stroke and at the beginning of the expansion stroke.

I declare that all statements made herein of my own knowledge are true and that all statements made on information and belief are believed to be true, and further, that these statements were made with the knowledge that willful false statements and the

like so made are punishable by fine or imprisonment, or both, under Section 1001 of Title 18 of the United States Code, and that such willful false statements may jeopardize the validity of U.S. Patent No. 6,688,280.

Dated: 1/20/06

By: ____________________
Joel Hiltner

EXHIBIT A

Curriculum Vitae

Dr. Joel Hiltner

Education:

1992	B.S.M.E. (Summa Cum Laude) from The Ohio State University
1993	Master's degree in Mechanical Engineering from The Ohio State University
1997	Ph.D. in Mechanical Engineering from The Ohio State University

Work Experience:

1991 – 1997	Graduate research fellow in the Department of Mechanical Engineering at The Ohio State University. Research topic for Master's and Ph.D. programs involved optical diagnostics in internal combustion engines. Primary researcher on projects funded by Honda R&D of Japan and several U.S. Tier 1 automotive suppliers. Graduate studies supported by a National Science Foundation Fellowship, Ohio State University Presidential Scholars Program, and yearly grants from Honda of America.
May 1997 – July 1998	Caterpillar Engine Research, Alternative Fuels Group, Peoria Illinois. Work included thermodynamic and fluid dynamic modeling, testing, and design of internal combustion engines for heavy duty mobile and stationary applications.
Sept. 1998 – Sept. 1999	Post-doctoral research fellow at Trinity College, Dublin, Ireland. Primary researcher on European Union funded project aimed at minimizing emissions from pulse combustors used for stationary power generation. Designed and constructed optical test combustor as well as carrying out detailed chemical kinetic and fluid dynamic modeling.
Sept. 1999 – Feb. 2000	Visiting research fellow at University of California at Berkeley. Worked with team of researchers developing Homogeneous Charge Compression Ignition engine for high efficiency, low NO_x operation. Project included cycle simulation as well as development of engine test facility and extensive engine testing.
March 2000 – Jan. 2001	Visiting fellow, RMIT University, Melbourne, Australia. Developed a senior level class on engines and transmissions to support new Automotive Engineering degree at RMIT. Worked with local automotive industry (Ford of Australia, GM Holden) to develop engine test cell for future research and student instruction. Created web enabled course content to allow distance learning by students worldwide through RMIT.
Aug. 1998 – Present	Chief engineer for Hiltner Combustion Systems. Work focused on efficiency improvement and emissions reduction through combustion system improvement for all varieties of combustion engines. Efforts include modeling of non-traditional engines and engine cycles, optimization of spark ignited natural gas engines for stationary power generation, and the development of experimental and analysis tools for engine test cell applications.

A Brief Description of Hiltner Combustion Systems:

Hiltner Combustion Systems, of which I am the sole-proprietor, works with a range of clients in order to develop clean, efficient, and profitable engines for a number of worldwide markets. These clients include OEM engine manufacturers in the automotive and heavy duty markets, control and fuel system suppliers, university research teams, and small start up companies. Simulation tools developed by Hiltner Combustion Systems are in use at a number of firms worldwide for the development and refinement of heavy duty engines. A representative list of clients I have assisted in my role at Hiltner Combustion Systems over the last five years includes: Woodward Governor (Ft. Collins, CO), Caterpillar Inc. (Peoria, IL), Fiberdynamics Inc. (Bryan, TX), Perkins Engine Company (Stafford, U.K.) (affiliated with Caterpillar Inc.), MaK (Kiel, Germany) (affiliated with Caterpillar Inc.), and Atlas Engine Company (Boulder, CO), as well as a number of other entities in the automotive industry. My work for all of these clients has pertained primarily to thermodynamics, combustion chemistry, engine controls, and hardware development for spark ignited and diesel power plants. In my work at Hiltner Combustion Systems, I have played a major role in at least three engine families that are now in series production by major manufacturers.

My role at Hiltner Combustion Systems has also involved collaboration with a number of universities and National Laboratory researchers on advanced research and development topics focused on low emissions combustion technologies. These projects have included HCCI engine development with Lawrence Livermore National Laboratory and UC Berkeley, NO_x aftertreatment technology with The Ohio State University, and government grant reviews for the California Energy Commission and the U.S. Department of Energy. These research activities are augmented by my active participation in engine related technical organizations. My extensive work with SAE (Society of Automotive Engineers), ASME (American Society of Mechanical Engineers), The Combustion Institute, and CIMAC (Conseil International des Machines a Combustion) has included the publication of several papers, organization of a number of conference sessions, and extensive review of technical publications for conferences and journals.

Selected Publications:

Hiltner, J.D., and Kennedy, L.A., "The Development, Fabrication, and Testing of an Optical Access Engine for Internal Combustion Engine Research", The Combustion Institute, Central States Section, 1994 Technical Meeting, June 5, 1994, pages 412-417.

Hiltner, J.D., and Samimy, M., "A Study of In-Cylinder Mixing in a Natural Gas Powered Engine by Planar Laser Induced Fluorescence", SAE Paper Number 961102, 1996.

Hiltner, J.D., and Samimy, M., "The Impact of Injection Timing on Mixture Formation in a Natural Gas Powered Engine", SAE Paper Number 971708, 1997

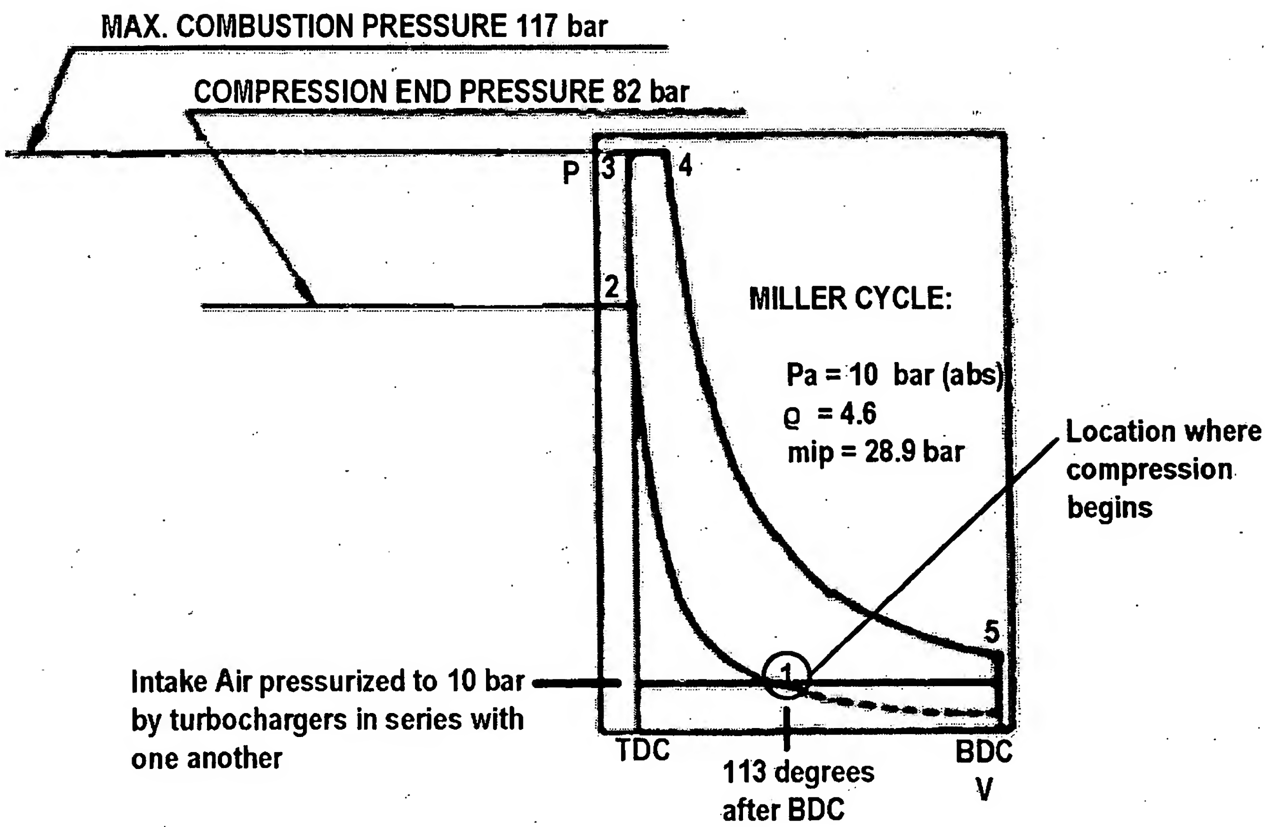
Hiltner, J.D., "The Impact of Fuel Distribution on Cyclic Combustion Variations in a Natural Gas Fuelled, Spark Ignition Engine", Ph.D. Dissertation, The Ohio State University, 1997.

Hiltner, J.D., Fiveland, S.B., Willi, M.L., Agama J.R., "System Efficiency Issues for Natural Gas Fueled HCCI Engines in Heavy-Duty Stationary Applications", SAE Paper #2002-01-0417.

Hiltner, J.D., Mauss, F., Johansson, B., Agama, J.R., "HCCI Operation with Natural Gas: Fuel Composition Implications", ASME Journal of Engineering for Gas Turbines and Power, July 2003, pages 837-844.

Hiltner, J.D., Fiveland, S.B., "Development Considerations for Lean Burn Natural Gas Engines Employing the Miller Cycle", 24th CIMAC World Congress on Combustion Engine Technology, Kyoto, June 2004

Exhibit B
Annotated Fig. 6 from Zappa et al.



$t_1 = 90°C$
$t_2 = 544°C$
$t_3 = 844°C$
$t_4 = 1712°C$
$t_5 = 985°C$

EXHIBIT C

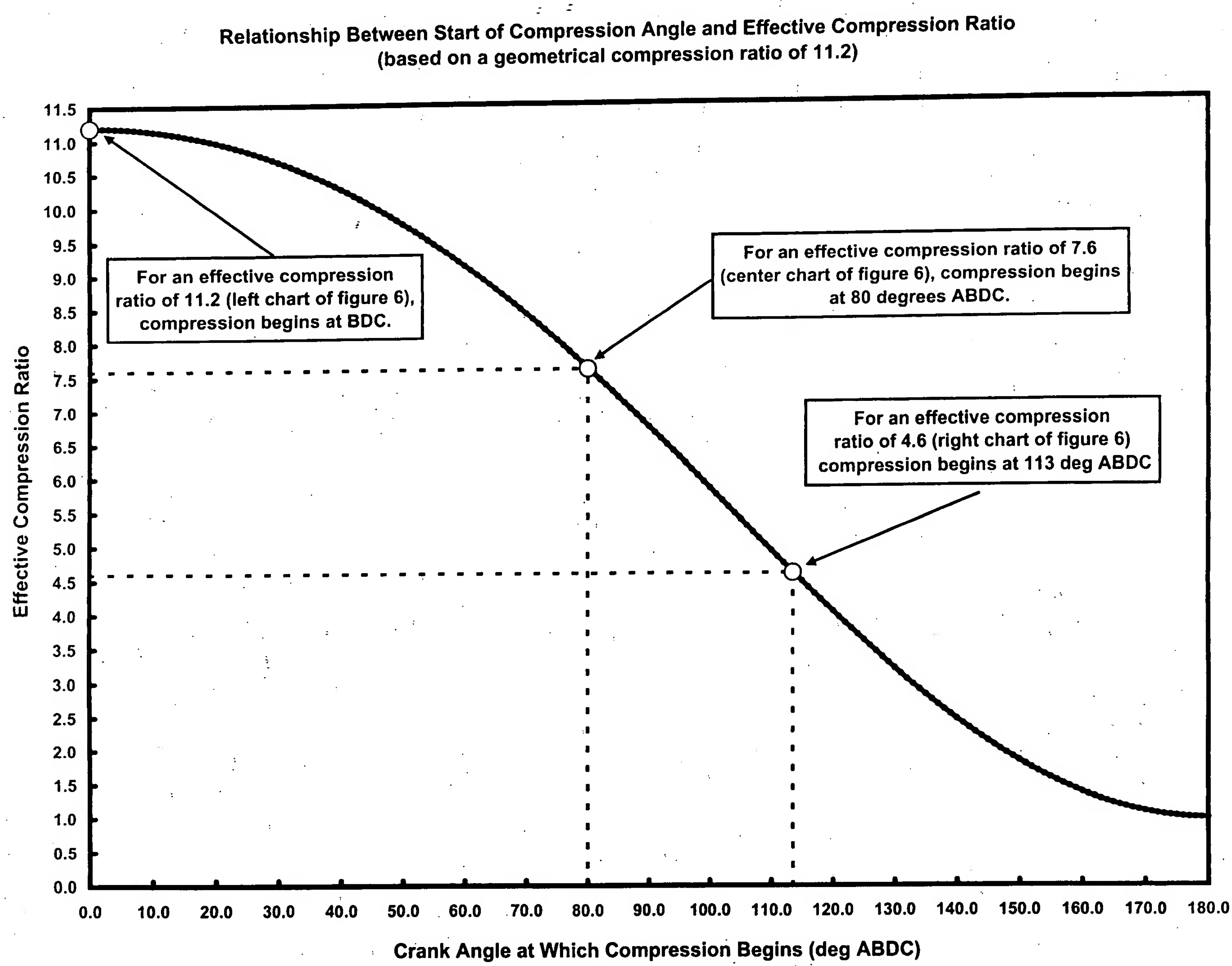

Relationship Between Start of Compression Angle and Effective Compression Ratio
(based on a geometrical compression ratio of 11.2)
Effective Compression Ratio
Crank Angle at Which Compression Begins (deg ABDC)
For an effective compression ratio of 11.2 (left chart of figure 6), compression begins at BDC.
For an effective compression ratio of 7.6 (center chart of figure 6), compression begins at 80 degrees ABDC.
For an effective compression ratio of 4.6 (right chart of figure 6) compression begins at 113 deg ABDC

Exhibit D
Annotated Fig. 8 from Zappa et al.

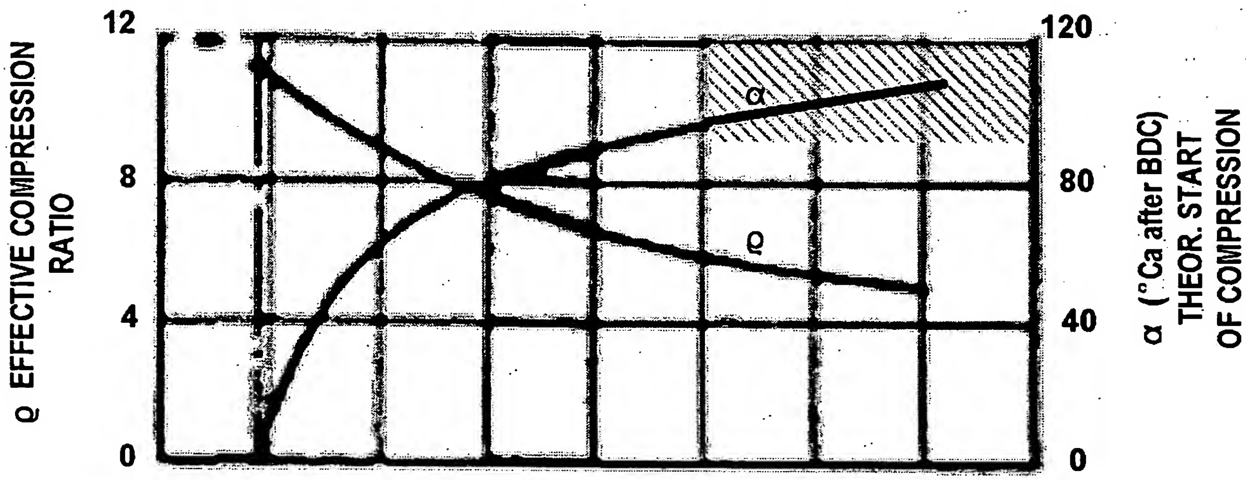


EXHIBIT E

INTERNAL COMBUSTION ENGINES *Analysis and Practice*

BY EDWARD F. OBERT PROFESSOR OF MECHANICAL ENGINEERING AT THE TECHNOLOGICAL INSTITUTE OF NORTHWESTERN UNIVERSITY

SECOND EDITION

SCRANTON, PENNSYLVANIA
INTERNATIONAL TEXTBOOK COMPANY

Printed in the United States of America

First Edition
First Printing, March, 1944
Second Printing, August, 1945
Third Printing, December, 1946
Fourth Printing, March, 1947
Fifth Printing, January, 1948
Sixth Printing, July, 1948
Seventh Printing, December, 1948
Eighth Printing, August, 1949
Ninth Printing, November, 1949

Second Edition
First Printing, September, 1950
Second Printing, November, 1950
Third Printing, July, 1951

The Haddon Craftsmen, Inc.
Scranton, Pennsylvania

1438

At full throttle, the pressures during the intake stroke should be close to atmospheric for an unsupercharged engine. When fluid friction is present and therefore the pressures are below atmospheric, *two* losses will arise: (1) The pumping loss will be increased and (2) the weight of mixture inducted into the engine will be decreased. Since the indicated work of the engine is directly proportional to the weight of mixture inducted, friction losses on the intake stroke are more serious than losses of similar magnitude on the exhaust stroke. For this reason, the intake valve and port, rather than the exhaust valve and port, should be made as large as possible.

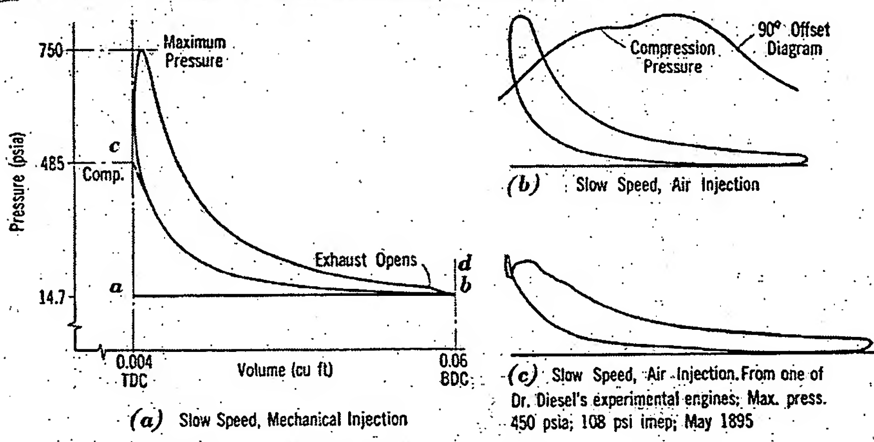


FIG. 5-11. *pV* diagrams for CI engines at full load.

Since no provisions are made in the usual engine for either adjustable cams or camshaft timing, and since time is an important factor in either induction or exhaust, it should be apparent that engine speed will affect the pressures in the engine. As the design speed of the engine is raised, in general, the valves should be opened earlier and closed later in the cycle if the negative work area is to be kept small (impractical).

5-9. Pressure Diagrams for the CI Engine. The *pV* diagram for the four-stroke-cycle CI engine is quite similar to that for the SI engine (compare Fig. 5-8 with Fig. 5-11*a*). When a blast of air is used for injecting and atomizing the fuel, the maximum pressure can be held to a low value and combustion can occur at approximately constant pressure (Figs. 5-11*b* and *c*). The combustion period can be expanded on the diagram by operating the indicator out of phase with the engine. In Fig. 5-11*b*, a 90-deg offset diagram is shown. Here the reciprocating drum of the indicator was driven by a crank which was displaced 90 deg from the engine crankshaft.

When the *pt* diagram is studied, certain characteristics of the CI engine become apparent. In Fig. 5–12, four stages (Art. 4–12) of diesel combustion are illustrated:

1. Ignition delay
2. Uncontrolled combustion
3. Controlled combustion
4. Late burning, or afterburning

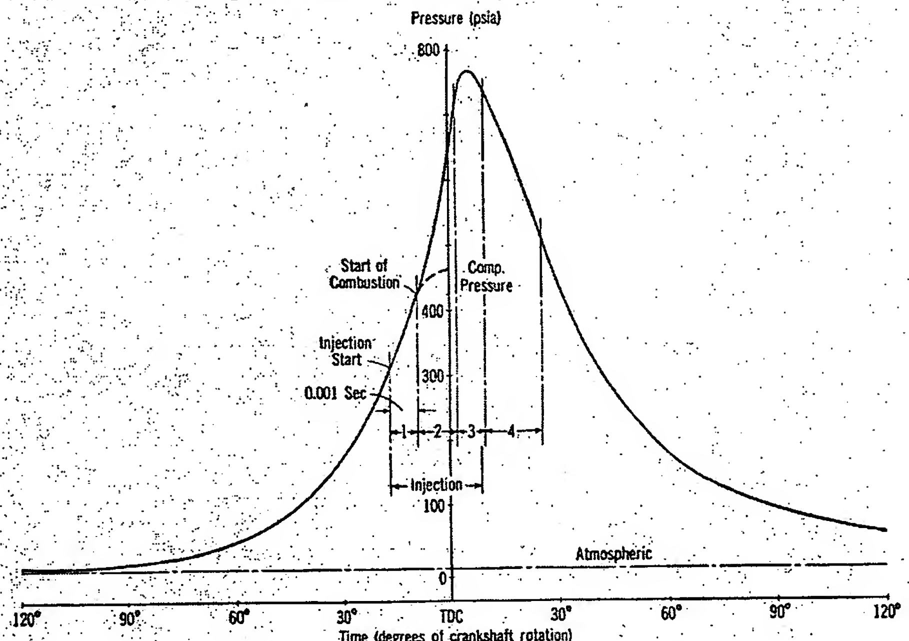


FIG. 5–12. *pt* diagram for mechanical-injection CI engine at full load.

If the ignition-delay period of the fuel is long, a relatively large amount of fuel will be injected and will accumulate in the engine, and the second stage of combustion may be particularly violent, with high rate of pressure rise. On the other hand, with a short delay period compared to the duration of injection, the second stage can be somewhat suppressed because a part of the fuel is injected into very hot and burning gases and therefore combustion and injection may proceed in unison. Of course, the pressures and rates in the engine depend on many factors,[13] such as (1) the timing of the start of injection, (2) the ignition-delay time of the fuel, (3) the speed of the engine, and (4) the duration of injection. In Fig. 5–11*c* it can be seen that late injection allowed the pressure to fall on the expansion stroke before combustion occurred,

[13] Fig. 15–39.

with the maximum pressure being lower than the compression pressure. With short ignition delay, long duration of injection, low speed, and injection beginning near TDC, Diesel's principle of constant-pressure combustion can be approached.

In most CI engines it is desirable to burn the fuel quickly near TDC in order to obtain low fuel consumption (since energy liberated at TDC is available to do work throughout the expansion stroke). The injection timing and duration are adjusted to give pV diagrams similar to Fig. 5–11*a*. Thus the usual CI engine operates closer to the Otto cycle than to the Diesel cycle.

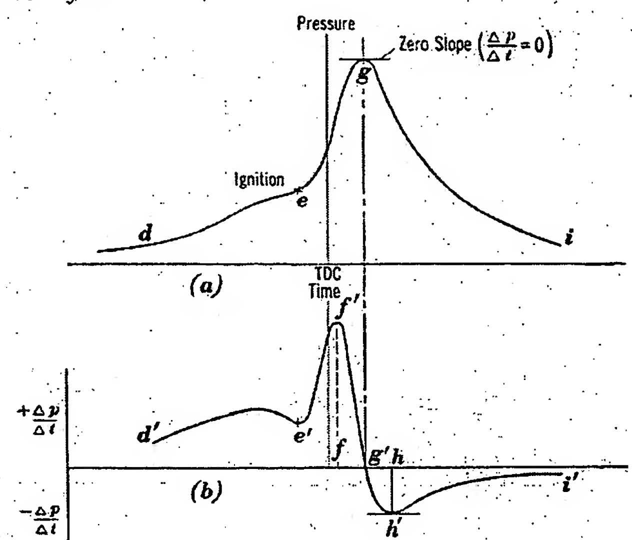


FIG. 5–13. *pt* and rate diagrams for nondetonating combustion.

The CI engine tends towards roughness because the initial pressure rise is abrupt. Autoignition results in not only high rates of pressure rise, but also extremely high accelerations (Art. 5–7) of the rate. Since maximum pressures are also high, the usual CI engine must be heavily constructed.

When the CI engine is operated at part load, the quantity of fuel is decreased, but not the quantity of air. For this reason the pressures on the compression stroke of Fig. 5–11 (or Fig. 5–12) are not affected although the pressures on the expansion stroke are reduced and the over-all expansion line is lowered on the diagram as the load is decreased. Since the air intake was not throttled, the negative work area is small. Thus the CI engine has a more efficient method of control than the SI engine.

$$\eta_t = \frac{Q_A + Q_R}{Q_A} = 1 - \left(\frac{1}{k}\right)\left(\frac{T_d - T_a}{T_c - T_b}\right) = 1 - \left(\frac{1}{k}\right)\frac{T_a\left(\frac{T_d}{T_a} - 1\right)}{T_b\left(\frac{T_c}{T_b} - 1\right)} \quad (a)$$

Since $\Delta s_{ad} = \Delta s_{bc}$ in Fig. 6–3b

$$\Delta s_{ad \text{ or } bc} = c_v \ln \frac{T_d}{T_a} = c_p \ln \frac{T_c}{T_b} \quad \text{or} \quad \left(\frac{T_c}{T_b}\right)^k = \left(\frac{T_d}{T_a}\right) \quad (b)$$

And by Eq. (3–27)

$$\frac{T_a}{T_b} = \left(\frac{v_b}{v_a}\right)^{k-1} = \frac{1}{r_v^{k-1}} \quad (c)$$

Calling T_c/T_b the load ratio L, and substituting Eqs. (b) and (c) in Eq. (a),

$$\eta_t = 1 - \frac{1}{r_v^{k-1}}\left[\frac{L^k - 1}{k(L-1)}\right] \quad (6\text{–}5)$$

Note that Eq. (6–5) for the Diesel cycle differs from Eq. (6–1) for the Otto cycle only by the bracketed term, which is always greater than unity. Thus the efficiency of the Diesel cycle is less than the efficiency of the Otto cycle when comparison is made at the same expansion ratio and for the same working medium. Although the Otto-cycle efficiency was independent of load, the Diesel-cycle efficiency progressively increases as the load is decreased (and equals that of the Otto cycle at the limit of zero load).

6–3. The Dual Cycle. In modern CI engines the pressure is not constant during the combustion process, but varies in the manner illus-

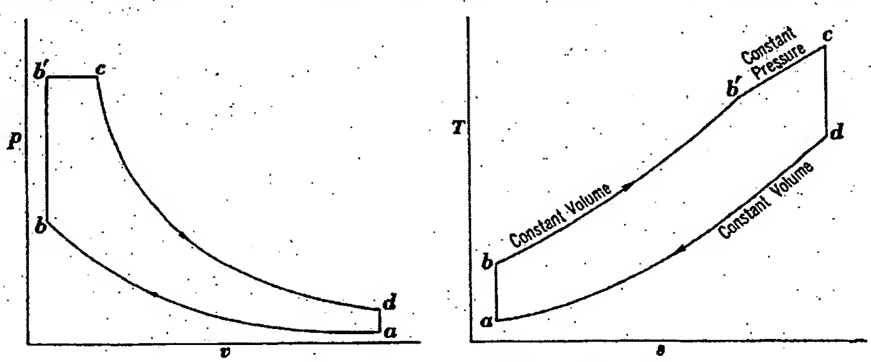


FIG. 6–4. Air-standard Dual cycle.

trated in Fig. 5–12. Here the initial part of the combustion period occurs at approximately constant volume while the final part will be considered to occur at approximately constant pressure. A hypothetical cycle with the following processes can be premised for this engine (Fig 6–4):

ab, isentropic compression
bb′, constant-volume addition of heat
b′c, constant-pressure addition of heat
cd, isentropic expansion
da, constant-volume rejection of heat

Discussion of this cycle will be reserved for Art. 6–4. Note that the modern diesel engine can have continued burning on the expansion stroke (Art. 5–9) and, like the Dual cycle, does not liberate all the energy at constant volume.

6–4. Comparison of Air-standard Cycles. For any given expansion ratio and given heat input, the thermal efficiency is highest for the Otto cycle and decreases in the following order:

1. Otto cycle
2. Dual cycle
3. Diesel cycle

The Otto cycle allows the most complete expansion and attains the highest efficiency because all the heat is added before the expansion process is under way. The Diesel cycle is the worst in this respect, since the last portion of the heat is supplied to fluid that has a relatively short expansion before rejection occurs.

On the basis of the same heat input and the same maximum pressure, the order of efficiency is:

1. Diesel cycle
2. Dual cycle
3. Otto cycle

This comparison is important because the real diesel engine can use high compression ratios, while the SI engine is limited to relatively low ratios because of the restriction imposed by detonation (Art. 4–11).

Method of Graphical Construction. No attempt should be made to memorize the proofs in the following paragraphs, but the procedure in construction is of interest:

1. Determine the *two* restrictions in each comparison.
2. Sketch one of the cycles (say, the Otto cycle) on the *Ts* diagram.
3. Start from the initial state *a* of the Otto cycle and sketch in a Diesel cycle that obeys the given restrictions. (Remember here that lines of constant pressure are less steep at any *one* state on the *Ts* diagram than lines of constant volume.)
4. Set up an expression for thermal efficiency in terms of one of the quantities held constant in both cycles:

$$\eta_t = 1 - \frac{|Q_R|}{Q_A} = \frac{W}{Q_A} = \frac{W}{W + |Q_R|}$$

5. Determine from the *Ts* diagram the relative amounts of heat rejected and therefore which cycle is the more efficient.
6. Insert the Dual cycle between the two extremes: the Otto and Diesel cycles.

IDS Form PTO/SB/08: Substitute for form 1449A/PTO **INFORMATION DISCLOSURE STATEMENT BY APPLICANT** *(Use as many sheets as necessary)*				***Complete if Known***	
				Reexam Control Number	To Be Assigned
				Filing Date	January 20, 2006
				First Named Inventor	Weber (USP 6,688,280)
				Art Unit	To Be Assigned
				Examiner Name	To Be Assigned
Sheet	1	of	1	*Attorney Docket Number*	08351.0294-00

U.S. PATENTS AND PUBLISHED U.S. PATENT APPLICATIONS

Examiner Initials*	Cite No.[1]	Document Number Number-Kind Code[2] *(if known)*	Issue or Publication Date MM-DD-YYYY	Name of Patentee or Applicant of Cited Document	Pages, Columns, Lines, Where Relevant Passages or Relevant Figures Appear
		US-5,445,128	08/29/1995	Letang et al.	
		US-4,836,161	06/06/1989	Abthoff et al.	
		US-			
		US-			
		US-			
		US-			
		US-			
		US-			
		US-			
		US-			
		US-			

Note: Submission of copies of U.S. Patents and published U.S. Patent Applications is not required.

FOREIGN PATENT DOCUMENTS

Examiner Initials*	Cite No.[1]	Foreign Patent Document Country Code[3] Number[4] Kind Code[5] *(if known)*	Publication Date MM-DD-YYYY	Name of Patentee or Applicant of Cited Document	Pages, Columns, Lines, Where Relevant Passages or Relevant Figures Appear	Translation[6]

NON PATENT LITERATURE DOCUMENTS

Examiner Initials*	Cite No.[1]	Include name of the author (in CAPITAL LETTERS), title of the article (when appropriate), title of the item (book, magazine, journal, serial, symposium, catalog, etc.), date, page(s), volume-issue number(s), publisher, city and/or country where published.	Translation[6]
		Zappa, G. and Franca, T., "A-4-Stroke High Speed Diesel Engine With Two-Stage of Supercharging and Variable Compression Ratio," 13th International Congress on Combustion Engines (CIMAC), 1979, pages D19-1 - D19-22.	

Examiner Signature		Date Considered	

EXAMINER: Initial if reference considered, whether or not citation is in conformance with MPEP 609. Draw line through citation if not in conformance and not considered. Include copy of this form with next communication to applicant.

13e CONGRES INTERNATIONAL DES MACHINES A COMBUSTION

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A 4—STROKE HIGH SPEED DIESEL ENGINE WITH TWO—STAGE OF SUPERCHARGING AND VARIABLE COMPRESSION RATIO

by

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D 19

A 4-STROKE HIGH SPEED DIESEL ENGINE WITH TWO-STAGE OF SUPERCHARGING AND VARIABLE COMPRESSION RATIO

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SUMMARY

The Authors' Company has introduced a new engine called B 230 DV that can develop a 25 bar mean effective pressure.

The main technical innovations are represented by the double stage of supercharging and the variable compression ratio.

The 20-cylinder version of the B 230 DV engines is being as sembled, after the operational characteristics have been checked on an 8-cylinder experimental prototype.

The paper describes shortly the studies and experience which concern this engine.

RESUME

La Compagnie des Auteurs a mis au point un nouveau moteur, appelé B 230 DV, capable de développer une puissance moyenne effective de 25 bar.

Les principales innovations techniques sont représentées par le double étage de suralimentation et le taux de compression varia ble.

Aprés avoir vérifié les caractéristiques fonctionelles sur un prototype expérimental à 8 cyl. on a en cours de réalisation la version B 230.20 DV à 20 cyl.

Ce mémoire décrit brièvement toutes les études et les expériences concernant ce moteur.

1. THE B 230.20 DV ENGINE

Presentation

The B 230 DV is a new engine with high specific outputs and characterized by some unconventional features.

The geometrical characteristics are:

- cylinder bore 230 mm
- cylinder stroke 270 mm
- cylinder number 20, vee-form

The performances are the following:

- power 5,600 kW
- rotational speed 1,200 r.p.m.
- m.e.p. 25 bar

12 engines are being built, two of which in the assembly stage.

Fig. 1 shows the external views. The B 230.20 DV engine is derived from the conventional B 230.20 engine (see photo of Fig. 2) which has a rated power of 4,000 kW and which continues to exist beside the new version. The power increase is 40% and it is reached by the sole increase of m.e.p.. Weight and overall dimensions of the two engines are not much different and the weight per unit output decreases from 5.8 to 5.1 kg/kW.

The main features which characterize the B 230.20 DV engines are:

- double stage of supercharging and air cooling;
- variable compression ratio.

Other variants of smaller entity are:

- increase of the injection equipment (to face the increased deliveries of fuel required);
- increase of some components of the timing system, springs, tappets, cams (to face the increased pressures of the supercharging system).

The other engine components (frame, crankshaft, cyl. heads, liners, pistons, connecting rods, valves, bearings) are unchanged granting a large interchangeability of the components between the two versions. Also pumps, oil and water coolers remain unchanged, with the exception, obviously, of the diesel oil feeding pump.

Description

- Two-stage Supercharging System.

The supercharging system consists of two modules, installed on the engine: each includes two turbochargers and two air coolers. Feeding of the high pressure turbine is of multi-pulse type.

The four air coolers are arranged side-by-side within the vee; the low pressure ones at the ends and the high pressure ones in the middle. The two coolers of each module form a compact block with only one water inlet and one water outlet.

Air flows upwards through the low pressure cooler and subsequently downwards through the high pressure cooler.

Water follows a U-shape course passing in series from the low pressure cooler and the high pressure in the in-going passage and viceversa in the come-back passage.

- Variable Compression Ratio.

The variation of the compression ratio is made virtually by changing the timing of the intake valves. An advance of the intake valve closing gives way to an effective compression ratio smaller than the geometrical ratio of the engine, as it will be seen in details in the chapter: "Miller Cycle - Influence of the Intake Valve Closing on the Engine Effective Compression Ratio". The device which performs this variation is schematically shown in Fig. 3.

The control gear is made up by the cam, the roller, the shoe and the valve tappet. Roller and shoe are both carried by levers. Timing variation is performed through rotation of the excentric, fulcrum of the roller-bearer lever.

Rotation of the excentric towards extreme positions causes the roller to move in the sense of cam rotation or viceversa, thus delaying or anticipating the lifting phase; Fig. 3a refers to the conditions of max compression ratio and Fig. 3b to that of the min. compression ratio.

The max. angular variation of the valve lift phase is 40° crank angle.

2. CONSIDERATIONS ON THE B 230 DV PROJECT

Increase of Performances

An increase of output of the conventional engine performed with simple increase of m.e.p. within the same limits would imply remarkable increases either of the max. combustion pressure (20% about) and of the combustion chamber wall temperatures (on the average 50 to 100°C). These increases would be unbearable without important modifications of all the components.

The increase of the thermal stress can be avoided increasing the air charge pressure. The increase of mechanical stress can be avoided reducing the compression ratio in the cylinder. A combined solution can be therefore foreseen which enables an increase of performances to be attained, keeping unchanged both the stress level: mechanical and thermal. The typical features of the B 230DV engine (double stage and variable compression ratio) offer in fact these possibilities.

These features are not independent but correlated: supercharging pressure and compression ratio are chosen in order to attain the same compression pressure and the same max. pressure in the cylinder as in the standard engine; this means that the external air compression ratio (which takes place in the turbochargers) was increased by the same amount by which the compression ratio in the cylinder was decreased.

Prediction of Thermal and Mechanical Load

Working cycles are considered equivalent as to the thermal and mechanical load, if they give way, simultaneously:

- to the same max. combustion pressure;
- to the same temperature of the hot components.

The theoretical evaluations were based on "assimilated" cycles, Sabathé type, (with addition of the secondary cycle) suitably chosen to reproduce exactly the values of the significant engine thermodynamic parameters (m.i.p., air charge, weight, pressure and temperature, compression and max. combustion pressures, exhaust gas temperature considering also the scavenging air ratio and the efficiency of the turbocharger).

Temperatures variations of the hot components were evaluated by means of average parameters of thermal flow (effective temperature, heat transfer coefficient) which can be obtained from the cycle.

Efficiency Evaluation

Efficiency variations were evaluated as proportional to those of the ideal cycle efficiency, namely leaving out the "indicated" and mechanical efficiency.

This assumption can be considered valid at first approximation at least if the ratio air/fuel does not vary noticeably. In fact, Fig. 4 reports the measured efficiency curves of the conventional B 230 engine at 1,200 r.p.m.. The indicated efficiency curve decreases only slightly at high load simultaneously with the decrease of the ratio air fuel whereas the mechanical efficiency improves steadily.

Cycles with Reduced Compression Ratio

- Reduced Geometrical Compression Ratio

The simplest way to decrease the compression ratio is to increase the dead volume raising the cyl. head or lowering the piston. To attain an advantage on the thermal load it is necessary to associate the reduction of compression ratio with an increase of the air pressure. This condition is clarified in Fig. 5a for a constant m.i.p. of about 21 bar; it considers the temperature variation of a given point of the combustion chamber (starting from a pre-established reference condition of 350°C for $\varrho = 11.2$) with three possible assumptions.

The a) and b) curves assume constant supercharging with and without restoration of the max. pressure. The c) curve considers a reduction of ϱ with an increase of the air pressure such to restore the maximum and end compression pressures.

The mechanical load remains constant in a) and c) and decreases in b). The thermal load increases (slightly) in a), remains constant in b) and decreases strongly in c).

Fig. 5b shows the variations of the temperature considered in Fig. 5a versus m.i.p. and ϱ for the cycles modified according to assumption c).

The m.i.p. increases that can be obtained are:

for $\varrho = 10$ Δ m.i.p. = 15%

for $\varrho = 8.5$ Δ m.i.p. = 34%

- Cycle with Reduced Effective Compression Ratio

This cycle, as proposed by Miller, reduces the compression ratio through a reduction of the useful compression stroke ob-

tained in turn by the anticipated (or delayed) closing of the in take valves (see point "Miller Cycle - Influence of the Intake Valve Closing on the Engine Effective Compression Ratio").

The dead volume, as well as the expansion ratio, remains as in the conventional engine. Fig. 6 shows a comparison of the 3 cycles, equivalent as concerns thermal and mechanical load.

The first is a conventional cycle of 21 bar roughly, the others are Miller-type cycles with different supercharging pres sure (and effective compression ratio) having in common with the previous one the compression and the firing pressures. In Fig.7 is shown the variation of temperature of the reference point of the combustion chamber (350°C in the reference condition) for different m.i.p. and supercharging pressure with cycles achieved in the same way; it thus enables conditions of thermal equivalence to be immediately evaluated. Fig. 7 shows also the line of thermal equivalence calculated as suggested by Miller himself, that results sufficiently in agreement .

The m.i.p. that can be reached for the same wall temperature increases with the supercharging pressure, as a consequence of the increase of the air charge. However, some unfavourable effects are also met, as it can be gathered from Fig. 8:

- the efficiency of the cycle diminishes;
- the air/fuel ratio diminishes;
- the pressure drop between air and gas manifold pressure diminishes;
- the final compression temperature diminishes.

According to the indications of Fig. 8 the practical increase of the performances allowed by the Miller cycle should be limited to about 30%. This value corresponds to an air pressure of 5 to 6 bar (see the dark area of the figure) beyond which the values assumed by the compression ratio and by the end compression temperature can be considered unacceptable for a correct combustion.

Following the indications of Fig. 8, different solutions of increasing complexity can be envisaged to make use of the Miller cycle potential:

- Engine with a steady valve timing (the limit is given by the value of ρ sufficient to grant engine starting) Δ m.i.p. < 10%;
- engine with variable valve timing and single stage supercharging (the limit is given by the supercharging pressure: 4 to 4.5 abs. bar) Δm.i.p. = 10 to 20%;

- engine with a variable valve timing and double stage of supercharging (the limit is given by the decline of the engine thermodynamic parameters) Δm.i.p. $\cong$ 30%.

Comparison between Miller Cycle and Low (Geometrical) Compression Ratio Cycle

The comparison of the most significant parameters obtained with the two cycles is reported in Fig. 9 at parity conditions of the thermal and mechanical load.

The cycle with low compression ratio, at parity of air pressure allows higher m.i.p. to be achieved and has consequently higher development possibilities. It works, besides, with an increased air excess.

The Miller cycle, instead, is characterized by a better efficiency at parity of performances. This results from the higher expansion ratio utilized during a piston stroke.

The choice that the Authors' Company has made with the B 230 DV engine in favour of the Miller cycle with variable compression ratio was motivated not only by the better efficiency but also by the following considerations:

- simplicity and reliability of starting and light load operation, when compression ratio is that of a standard diesel engine;
- possibility of varying the valve timing to optimize cylinder volumetric efficiency in order to compensate for air lack during low speed operation.

Miller Cycle - Influence of the Intake Valve Closing on the Engine Effective Compression Ratio

The influence of valve timing on effective compression ratio is clarified by Fig. 10. The effective compression ratio is in practice influenced only by the intake valve closing and by the mean piston speed; it decreases, in comparison with the geometrical value, either for an anticipated closing and for a very delayed closing.

The effective compression ratio is, in each case, proportional to the geometrical compression ratio. A piston speed increase reduces the effective compression ratio in the cases of anticipated closing and increases it in the cases of retarded closing.

Variable Timing with Shifting of the Whole Intake Phase

The timing control device for the intake valve of the B 230 DV engine performs,in fact, not only a variation of the closing timing but also a shifting of the whole intake phase and, as a consequence, the variation of the compression ratio is associated with a variation of the valves opening angle and therefore with a variation of the overlap phase. This gives an additional advantage: in fact, at full load, with anticipated intake timing, the valve overlap is wide and allows a complete scavenging of the combustion chamber, whereas at low load, with retarded timing, the overlap is reduced and avoids or reduces the possible reverse scavenging.

This type of timing control keeps practically unaltered the total air flow through the engine and does not disturb the matching of the turbochargers; in fact the variation of the air trapped in the cylinder practically balances the variation of the scavenging delivery; see Fig. 11.

3. EXPERIMENTAL RESULTS

Experimental investigations have been carried out on an 8-cylinder vee-form engine, obtained with suitable modifications of the already existing standard version. Fig. 12 shows this engine at the test bench. The low pressure stage of the supercharging set is arranged for simplicity's sake on a separated framework Feeding of high pressure turbine takes place with the pulse converter system, namely with a system which is similar to that of the B 230.20 DV (multipulse).

The engine was instrumented to record and read all operation characteristic data. To measure the temperature several thermocouples have been installed on the combustion chamber and on the crank gear bearings. These measurements included pistons and exhaust valves.

Results of Setting-Up Tests

Both engine operating conditions, constant speed and propeller law, have been investigated and compared to the condition of the standard version.

Set-up included these main points:

- optimization of intake valve timing;
- matching of turbocharging system;
- optimization of combustion chamber and fuel injection.

- Operation at Constant Speed.

Fig. 13a shows engine characteristics, in final condition at constant speed (1,200 r.p.m.); from it the following considerations can be drawn:

- in comparison with the standard engine (dashed curves) it has been possible to increase the mean effective pressure from 17.7 to 24.5 bar with the same max. combustion pressure of 128 bar, with roughly the same turbine inlet gas temperature and with a slightly lower specific consumption: this means that the higher mechanical efficiency has more than balanced the reduction of the "ideal cycle efficiency";
- intake valve closing is kept constant until 15 bar m.e.p. and then gradually anticipated till 40° before BDC in order to keep the max. pressure within the pre-established limits of 128 bar. Air and gas pressure curves do not show any variation of trend and this confirms that timing variation does not disturb the turbochargers matching;
- a peculiarity can be observed instead on the curve of the gas temperature cylinder outlet which shows a max. relative value around 15 bar m.e.p., followed by a reduction between 15 and 20 m.e.p.. The same peculiarity does not appear on the curve of the gas temperature turbine inlet. This is, very likely, due to the fact that the thermocouple installed on the cyl. exhaust duct is strongly affected by the fresh air flow during overlap. This flow increases rapidly between 15 and 20 bar m.e.p. due to the increase of the overlap itself.

- Operation According to Propeller Law.

Fig. 13b represents the optimized operational characteristics according to propeller law (the max. power is, in this case, the one required for the service: 254 kW/cyl. at 1,240 r.p.m., 22 bar m.e.p.).

The optimization in this case led to the choice of a different camshaft with a larger intake phase and of a smaller closing advance (20° crank angle before BDC instead of 40°).

Intake timing varies continuously throughout the load range; again, the max. pressure reaches at the m.e.p. of 22 bar the set value of 128 bar.

The exhaust gas temperature at 1,200 r.p.m. (m.e.p. = 20.6 bar) is lower than that of the constant speed test (Fig. 13a) because of the larger intake phase.

The exhaust gas temperature curve has a form which is typic

al for operation according to propeller law. There is, in partic ular, a relative maximum of about 900 r.p.m. (≅ 12 bar m.e.p.) which, however, does not exceed the full load value.

Fig. 14 reports some significant comparisons of operation according to propeller law.

The reference (final) condition utilizes a camshaft with in take period of 230° crank angle; intake closing is constant until about 1,050 r.p.m. (12° after BDC, 108° overlap).

Fig. 14a shows the effect of a different timing (with the same camshaft) for speeds under 1,050 r.p.m.: for the earlier timing the exhaust gas temperatures and fuel consumption are much lower.

Fig. 14b shows the effect of a different intake period with the same closing angle, obtained with a different camshaft. The variant entails a reduction of the intake period of 20° at the costs of the overlap.

In the field of low and medium speeds the results match ap proximatively with these of Fig. 14a, whereas at high speed the values of the exhaust temperatures remain, as foreseeable, high er than the basic configuration.

Temperature of Engine Hot Components

The measurements have substantially confirmed, yet showing minor but significant deviations, the forecast that the temperatures of the hot components in B 230 DV engines would have not exceeded at m.e.p. of 24.5 bar the values obtained in the standard engine for a m.e.p. of 17.7 bar.

- Measurement at Constant Speed and Comparison with the B 230 Standard Engine

Fig. 15 shows this comparison for the different components at a constant speed of 1,200 r.p.m.. As concerns the temperatures of the cylinder head (Fig. 15a) the forecast proved to be right. As concerns the piston (Fig. 15b) the temperatures comply everywhere with the envisaged values, except for the periphery of the piston crown, where they result higher.

This is probably due to the fact that in the B 230 DV engine the fuel jets coming from the increased fuel injectors have an increased penetration and burn closer to the periphery of the combustion chamber.

Also on the liner, in fact (Fig. 15c) the temperatures are slightly higher than in the standard engine; they remain however within acceptable values.

Fig. 15d shows finally the temperatures pattern of the exhaust valve disc. In the B 230 DV engine both the temperature at full load and the variations of this temperature with the load are surprisingly low. This pattern is very likely due to the fact that as m.e.p. increases, the valves are more and more strongly cooled by the growing flow of scavenging air.

- Measurements According to the Propeller Law

Fig. 16 shows the temperatures of the hot components measured according to the propeller law for the two timing variants illustrated in Fig. 14a.

It can be seen that the remarkable exhaust temperature differences between the two variants correspond to significant although small differences, of the engine components.

At 1,200 r.p.m. (m.e.p. = 20.6 bar) the temperatures are generally lower than those reported in Fig. 15 for the same m.e.p. This, again, is due to the different camshaft selected.

It is worth of mention that, at least in the final configuration, the temperatures of the various components, do not exceed at any load the full load values.

Other Experimental Recordings

- Oscillographic Recordings Inside the Cylinder

Fig. 17 shows as an example the oscillographic recording, at a m.e.p. of 20.6 bar, of the pressures in the intake duct, inside the cylinder, and in the exhaust duct. Intake valves close 14° before BDC and overlap is 134°. The record proves the favourable scavenging condition during the whole overlap phase; one can also see the expansion with subsequent re-compression of the air charge in way of the BDC which is characteristic of the Miller cycle.

CONCLUSION

The whole series of tests has proved the favourable operating conditions of the 4-stroke engine with double stage of supercharging and variable compression ratio.

In particular, it has been confirmed the possibility of increasing by some 35 to 40% the output without exceeding the ther-

mal and mechanical stresses of the conventional engine and, as a consequence, of utilizing the same principal components of it.

A particular aspect that has been enlightened by the tests is the particular capability of this type of engine to operate according to propeller law which is, as well known, critical for engines with high degree of supercharging. This characteristic makes it particularly suitable for propelling sets as it is the case of the engines at present being built.

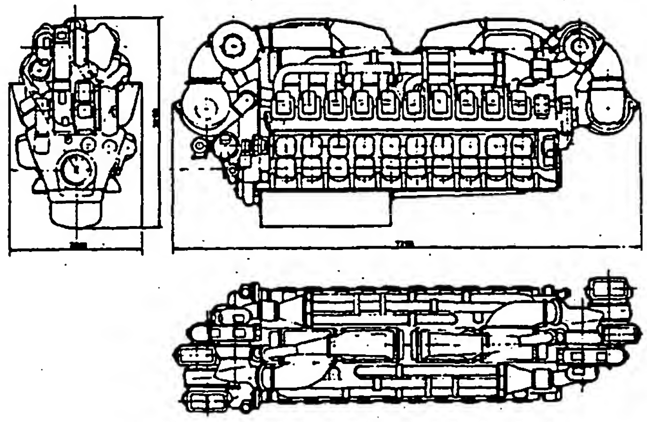

FIG. 1 – B 230.20 DV ENGINE – GENERAL VIEWS

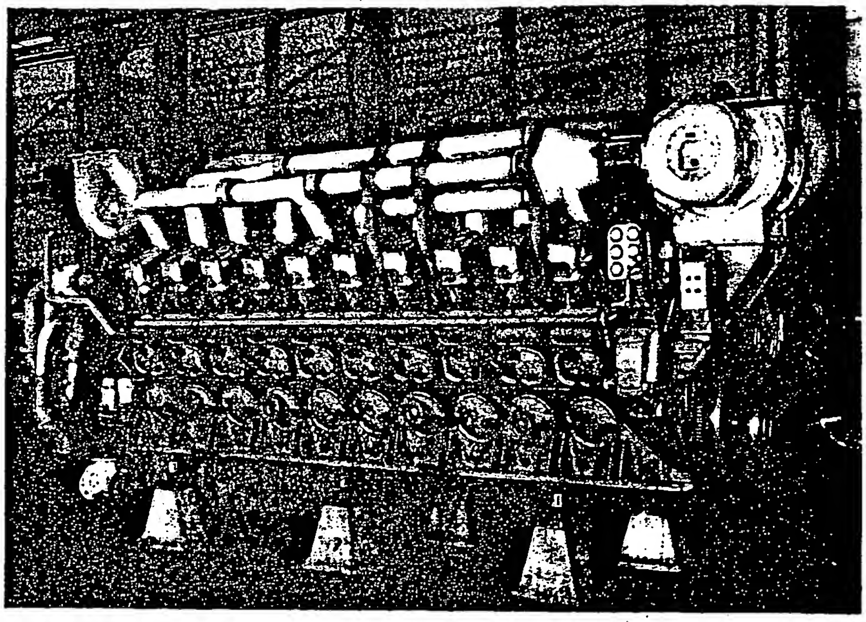

FIG. 2 8 230.20 ENGINE

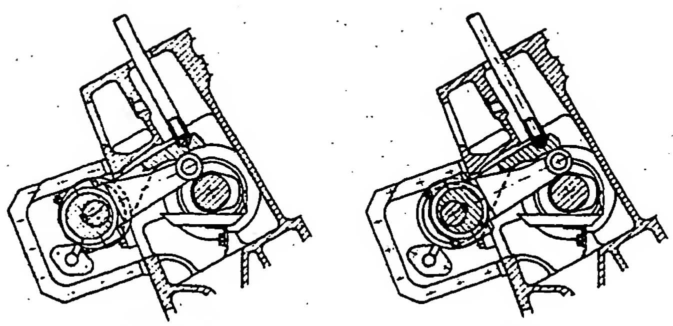

FIG. 3 B 230 DV ENGINE – INLET VALVE CONTROL DEVICE

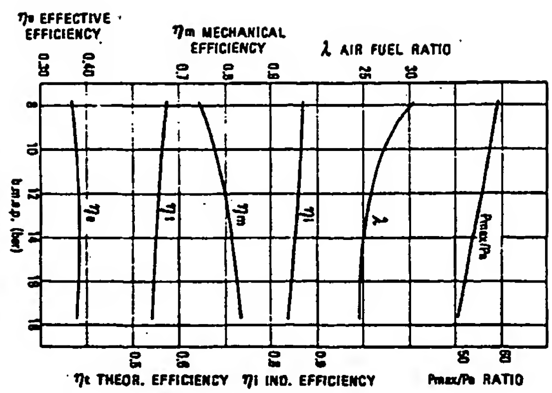

FIG.4 B 230 ENGINE – EFFICIENCY CURVES AT 1200 RPM.

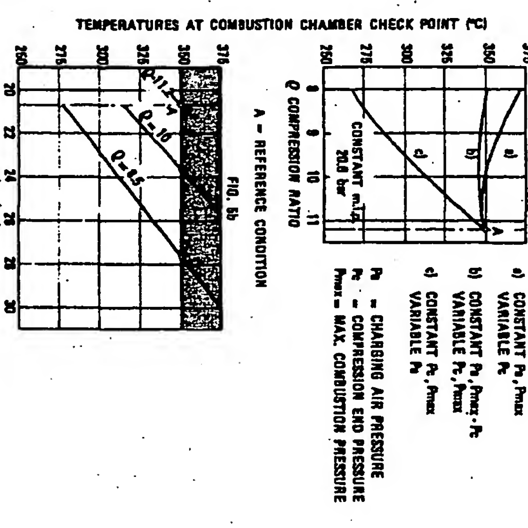

FIG.5 LOW GEOMETRICAL COMPRESSION RATIO CYCLE - COMBUSTION CHAMBER TEMPERATURES (CALCULATED VALUES).

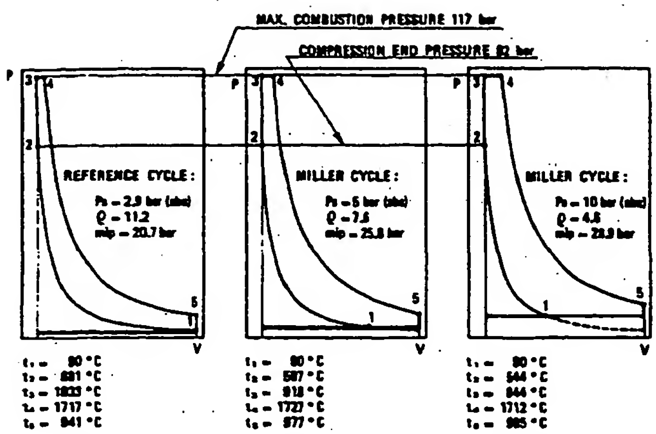

FIG.6 EQUIVALENT MILLER CYCLES WITH DIFFERENT CHARGING AIR PRESSURES

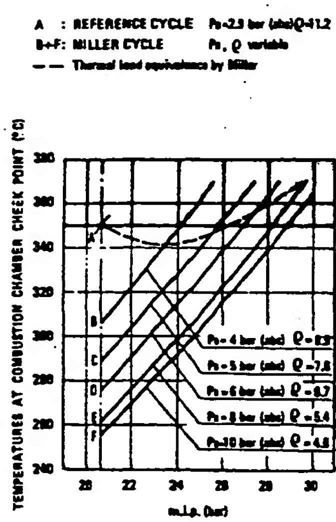

FIG.7
MILLER CYCLE – COMBUSTION CHAMBER TEMPERATURES (CALCULATED VALUES)

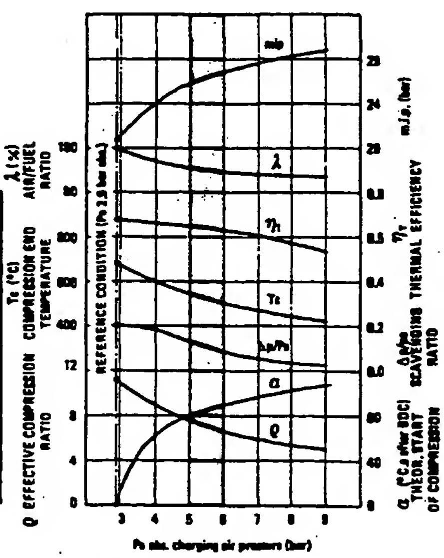

FIG.8
MILLER CYCLE – CHARACTERISTIC PARAMETERS VERSUS CHARGING AIR PRESSURE

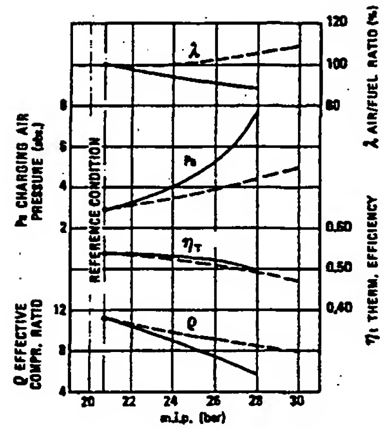

FIG. 9 CYCLE CHARACTERISTICS : COMPARISON BETWEEN MILLER AND LOW GEOMETRICAL COMPRESSION RATIO CYCLE.

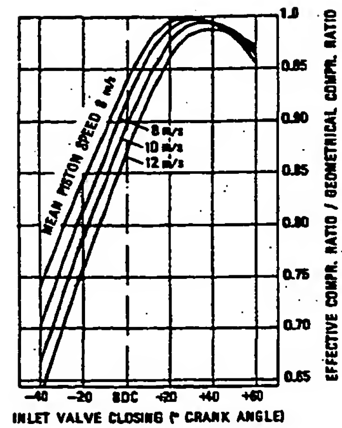

FIG. 10 EFFECTIVE COMPRESSION RATIO FOR MILLER CYCLE VERSUS INLET VALVE CLOSING.

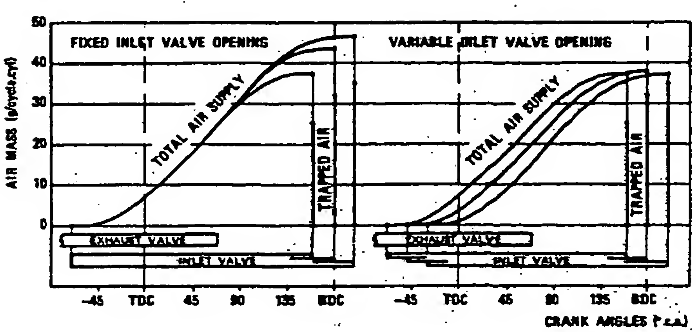

FIG. 11 CYLINDER AIR CAPACITY : INFLUENCE OF INLET VALVE TIMING. (B 230 ENGINE AT 1200 RPM, b.m.e.p. ~ 18 bar, Pa ~ 2.8 bar abs.).

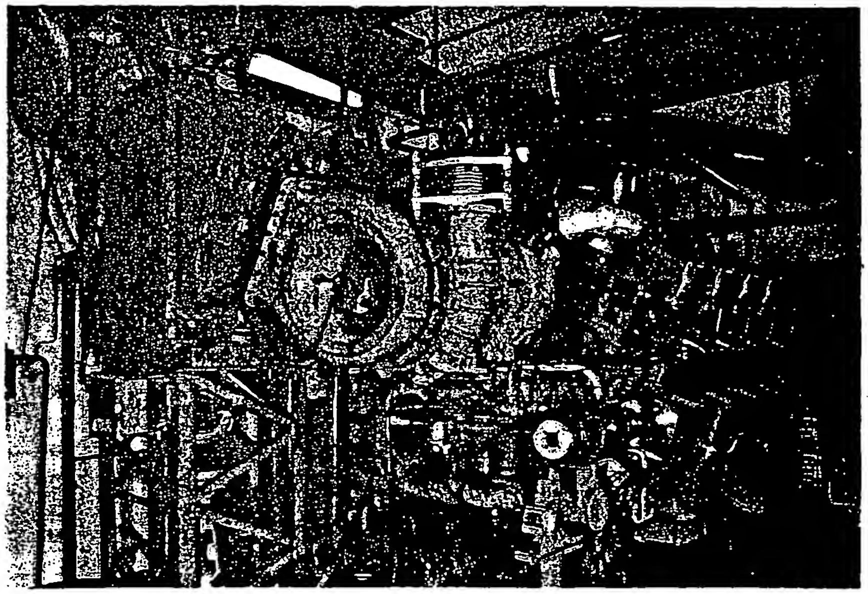

FIG. 12 – B 230.8 DV ENGINE AT THE TEST BENCH

FIG. 13a OPERATIONAL CHARACTERISTICS AT 1200 RPM

FIG. 13b OPERATIONAL CHARACTERISTICS FOLLOWING PROPELLER LAW

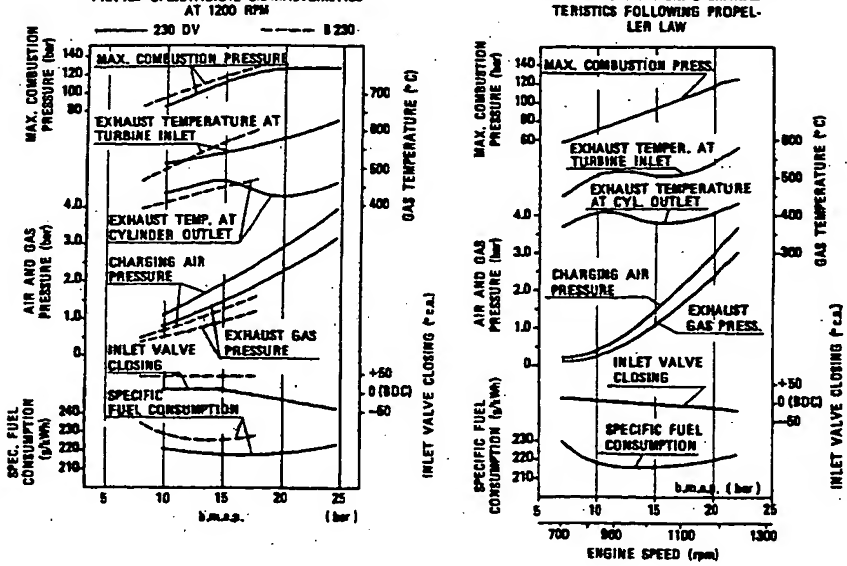

FIG. 13 B 230 DV ENGINE – OPERATIONAL CHARACTERISTICS

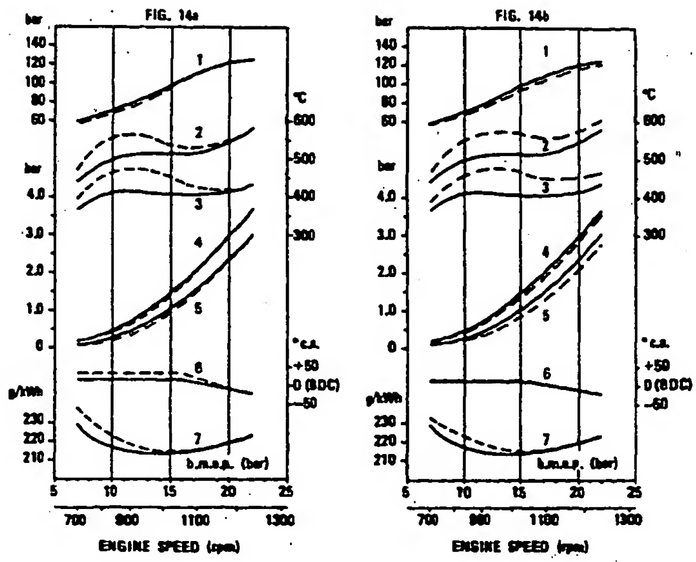

FIG. 14a INFLUENCE OF INLET VALVE TIMING (SAME CAMSHAFT) :

——— BASIC CONFIGURATION
– – – INLET VALVE TIMING DELAYED

FIG. 14b INFLUENCE OF INLET VALVE TIMING (DIFFERENT CAMSHAFTS)

——— BASIC CONFIGURATION
– – – OVERLAP DECREASED BY 20 c.a.; INLET VALVE OPENING DELAYED, CLOSING UNCHANGED

1 : MAX. COMBUSTION PRESSURE
2 : EXHAUST TEMPERATURE AT TURBINE INLET
3 : EXHAUST TEMPERATURE AT CYLINDER OUTLET
4 : CHARGING AIR PRESSURE
5 : EXHAUST GAS PRESSURE
6 : INLET VALVE CLOSING ANGLE
7 : SPECIFIC FUEL CONSUMPTION

FIG. 14 B 230 DV ENGINE — PERFORMANCE CURVES ACCORDING TO PROPELLER LAW. INFLUENCE OF VALVE TIMING

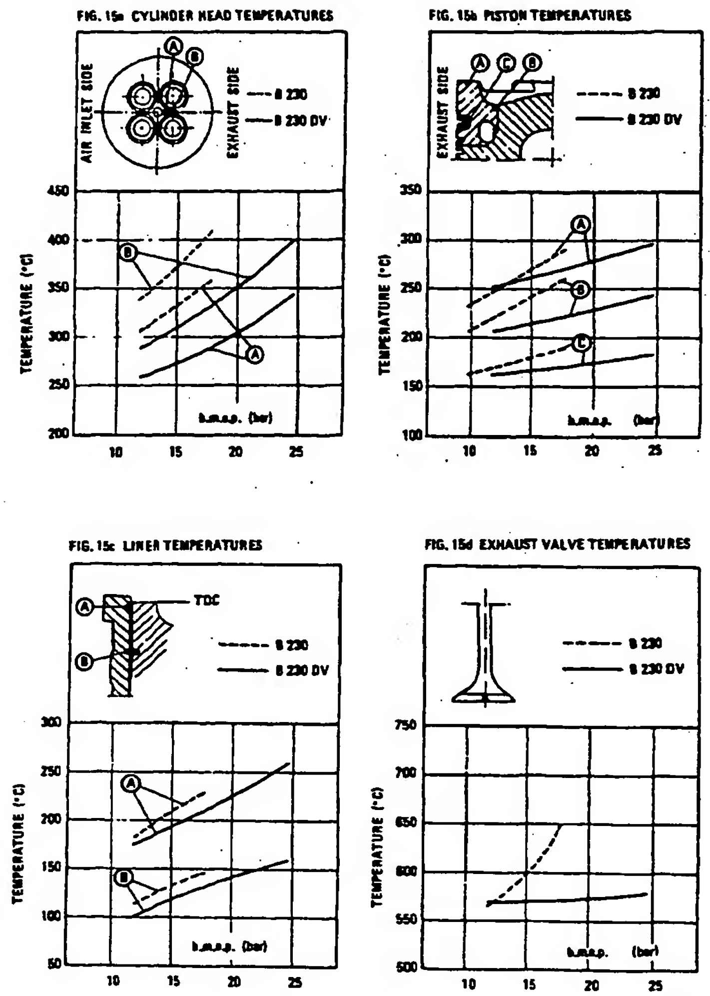

FIG. 15 COMBUSTION CHAMBER TEMPERATURE COMPARISON BETWEEN B 230 AND B 230 DV ENGINES AT 1,200 RPM

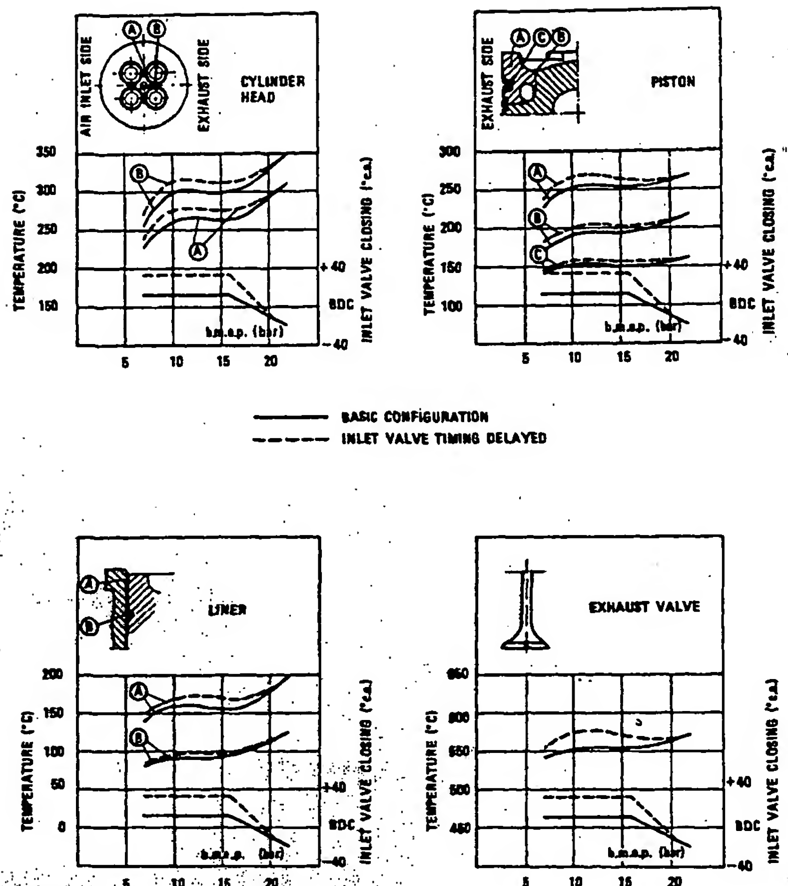

FIG. 18 B 230 DV ENGINE – COMBUSTION CHAMBER TEMPERATURES – PROPELLER LAW OPERATION WITH TWO DIFFERENT INLET VALVE TIMINGS.

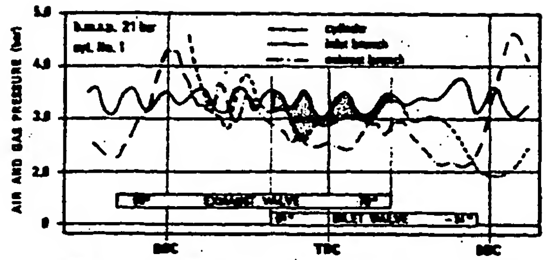

FIG. 17 B 230 8V ENGINE – EXHAUST GAS AND CHARGING AIR PRESSURE PATTERN.

UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 5,445,128

DATED : August 29, 1995

INVENTOR(S) : Letang et al.

It is certified that error appears in the above-indentified patent and that said Letters Patent is hereby corrected as shown below:

Col. 16, line 26, delete "max()" and insert --max ()--.

Col. 17, line 20, delete "LIM," and insert --LIM'--.

Col. 17, line 22, delete "max ()" and insert --max ()--.

Col. 17, line 23, delete "min ()" and insert --min ()--.

Col. 19, line 54, claim 1, delete "LIM," and insert --LIM'--.

Col. 19, line 54, claim 1, delete "(-$ACBr_{LIM}$" and insert --(-ACB_{LIM}--.

Col. 19, line 56, claim 1, delete "max ()" and insert --max ()--.

Col. 19, line 57, claim 1, delete "min ()" and insert --min ()--.

UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 5,445,128 Page 2 of 2

DATED : August 29, 1995

INVENTOR(S) : Letang, et al

It is certified that error appears in the above-indentified patent and that said Letters Patent is hereby corrected as shown below:

Col. 20, line 60, claim 1, delete "$(MAX_{TQ} - T_{TQLMT} - (D_{TQLMT} * TQ_{ADJ}))$" and insert --$(MAX_{TQ} - D_{TQLMT} - (D_{TQLMT} * TQ_{ADJ}))$--.

Signed and Sealed this

Twelfth Day of December, 1995

Attest:

Bruce Lehman

BRUCE LEHMAN

Attesting Officer

Commissioner of Patents and Trademarks

UNITED STATES PATENT AND TRADEMARK OFFICE
CERTIFICATE OF CORRECTION

PATENT NO. : 5,445,128
DATED : August 29, 1995
INVENTOR(S) : Dennis M. Letang et al.

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

Column 14,
Line 34, delete "T_{TQLMT}" and replace with -- D_{TQLMT} --.

Signed and Sealed this

Sixth Day of August, 2002

Attest:

JAMES E. ROGAN
Director of the United States Patent and Trademark Office

Attesting Officer